HEATING AND VENTILATING

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AIR CONDITIONING, REFRIGERATION, PIPING, AIR SANITATION, HEATING AND VENTILATION

Weave room of an air conditioned textile mill with supplementary humidifiers visible over duct outlets . . . page 57

Performance Index for Heat Pumps . page 63

Measurement and Storage of Solar Heat . . page 68

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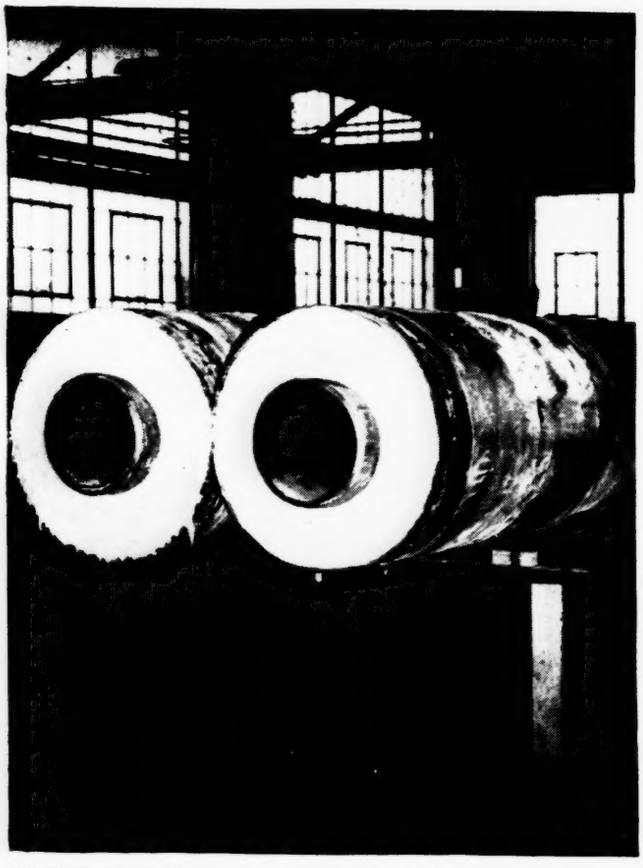
Minor Changes Unscramble School Air Supply . page 87

RIC-WIL PREFABRICATED UNITS FORM UNDERGROUND SECTIONS OF MILE-LONG TIE-LINE SUPPLYING 200,000 LBS. MORE STEAM PER HOUR WITH NO INCREASE IN BOILER EQUIPMENT



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VENTILATING

A MONTHLY MAGAZINE FOR ENGINEERS AND CONTRACTORS CONCERNED WITH THE DESIGN AND APPLICATION OF AIR CONDITIONING, REFRIGERATION, PIPING, AIR SANITATION, HEATING AND VENTILATION IN BUILDINGS

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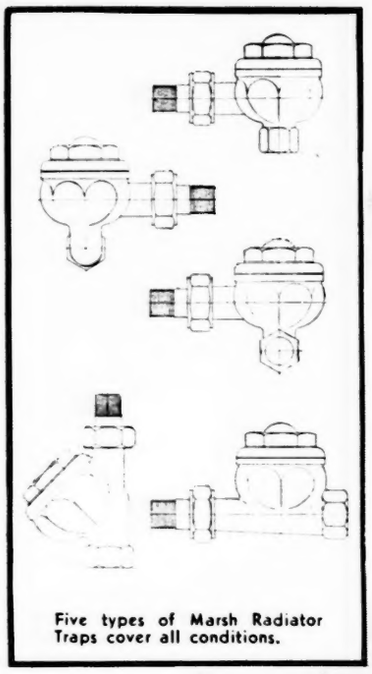
PUBLISHERS OF MACHINERY & MACHINERY'S HANDBOOK
The INDUSTRIAL PRESS

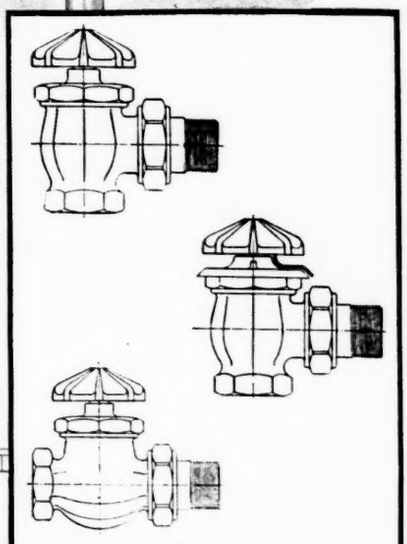
THIS MONTH'S COVER

Control of air conditions is very important to the successful operation of a textile mill. The cover photograph is one of the illustrations to accompany the article by P. L. Davidson on "Refrigeration for Textile Mill Air Conditioning."



Five types of Marsh Radiator Traps cover all conditions.

the Winning Combination . . . for those Vacuum Systems



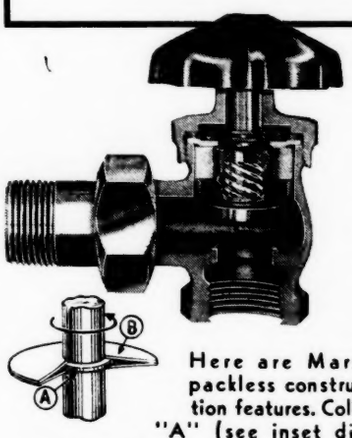
A type of packless valve for every service including graduated, extension stem, lock and shield type. Designed for pressures up to 125 lbs.

HERE is a team of thoroughbreds that will make every radiator function at its top efficiency in your steam or vapor systems—the combination that wins . . . against trouble, leakage, time.

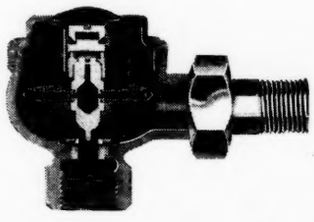
The trap is a masterpiece of precision—the kind that's always positive. Its distinguishing feature is a diaphragm highly sensitive to narrow differences in temperature between steam, water and air at operating levels. As a result of this fine differentiation, steam is retained until it gives its full heat, yet all condensate, air, and gases are positively passed into the return system. Note the excellent construction, attractive appearance and wide range of patterns shown in the accompanying illustrations.

Worthy teammate of the Marsh Radiator Trap is the packless radiator valve. In too many valves of this kind the word, "packless", is lightly used to describe a valve that merely requires less packing than the conventional packed valve, but this is not one of those cases. The Marsh has no packing whatever. Instead it has a metal-to-metal seal. It never blows out along the stem; never leaks; is practically indestructible. Note the diagram of this packless feature opposite. Note, also, the finer construction throughout.

Both trap and valve are individually tested under actual steam operation. They are typical of a full line of quality traps, vents and other heating specialties that always do their job right. Ask for complete facts about Marsh products and engineering service.



Here are Marsh packless construction features. Collar "A" (see inset diagram) is machined on stem. Stainless spring-steel wafer "B" is pressed against collar to form permanently leakproof annular seal. Cross section shows quality construction.



A highly sensitive diaphragm does the job in the Marsh Radiator Trap. Note sturdy construction.

MARSH HEATING EQUIPMENT COMPANY, 2059 Southport Avenue, Chicago 14, Ill.
Division of Jas. P. Marsh Corp.

MARSH SINCE 1865
Quality Heating Specialties

Washington News

LORING F. OVERMAN

LOTS of things of interest to the heating, ventilating and air-conditioning industries are implied by the recognition during April—in Washington and elsewhere—that price ceilings cannot be forgotten even though Chester Bowles and the OPA are only memories.

New Ceiling

The new "price ceiling" brought out into the open during April is not one that can be given one of the wartime MPR (Maximum Price Regulation) numbers. Instead it is an arbitrary ceiling fixed by a considerable number of economists, both in and out of Government service. This ceiling, they claim, is the likelihood that present prices of commodities and services are dangerously close to the possible peak of supply and demand. If the still-upward sweep can be arrested before the absolute peak is reached, the economists say, the expected slide down the other side will be less severe and less prolonged.

First official hint that economists believe prices and wages have reached the near-zenith came with a report to the President by his Council of Economic Advisors. Immediately the President launched, and all Government agencies are following up, a plea to business to reduce prices by 10%.

That Labor may also have taken the hint, although not specifically instructed by the President to do so, is implied by General Motors' completion of an agreement with CIO electrical unions calling for pay increases averaging 11%—considerably below original union demands. Conclusion in Washington is that this settlement will set the pattern for pending labor contracts in other major industries.

Modified Wage Demands?

If this assumption is sound, business can begin to plan on wage increases of 10 to 12%, instead of the 20 to 25% originally demanded by unions. Continuing the assumption, Business can also begin to figure its prices at levels somewhat lower than those previously considered necessary to cover anticipated increases in labor costs. If they can be kept at present levels, or reduced by the 10% requested by the President, the long-range result will be greatly improved, Washington believes.

As far as the building industry is concerned, "Continuation of high-cost trends may well result in pricing construction out of the market", in the opinion of Frank R. Creedon, Housing Expediter. "If restraint and sound judgment are exercised to stabilize building costs, the home-building industry should be able to carry through a sustained program of home-building in 1947—but at prices and cost levels higher than those prevailing in the 10 years preceding World War II.

He added, "It is possible that costs may have already reached their peak and a leveling may occur", but warned that an even greater flow of materials—together with limitations on non-housing construction —is needed to hold prices down.

Remove Controls

The Housing Expediter's claim that continued controls provide one of the primary answers to the housing problem is challenged by Frank W. Cortright, executive vice president of the National Association of Home Builders. Answering the expressed desire of the Administration that something be done to reduce construction costs, Mr. Cortright said that it can be accomplished by a full day's production by labor on the building site, by the reduced cost of building materials, and by the removal of the remaining government controls restricting the industry.

"The plain fact is that the price of houses has not risen to the same extent as the general price level—new houses are still a better buy than existing houses and most commodities and durable goods," Mr. Cortright said. "The construction cost of houses has increased only 68% since the 1935-39 period, during which time the cost of meat has gone up 99%, clothing 78% and food 83%. During the same period wages of workers in manufacturing industries went up 108%. Thus, the cost of new homes is not out of line with the ability of workers generally to pay for them.

"Of course, housing is caught in the inflationary spiral", Cortright said. "But this began with wage increases encouraged last year by the Administration leaders who now express concern with the resultant increase in the cost of construction."

In commenting on the statement of Housing Expediter Creedon regarding the housing situation Mr. Cortright endorsed the proposal that labor, manufacturers of building materials and home builders cooperate to stop the present dangerous upward trend in building costs. He said: "The country in desperate for rental housing and the industry can readily meet this need but it must be permitted to operate to its maximum capacity—freed of government domination."

Need for Rental Housing

About the only point on which there is any agreement between the Administration's housing policy and that of private enterprise is the recognition of a need for rental housing in almost unlimited amounts. In a new booklet, "Rental Housing", issued by the Office of the Housing Expediter for distribution to investors, builders and financial institutions, Mr. Creedon and Raymond M. Foley, National Housing Administrator, join in the following statement: "We sincerely believe that opportunities for developement of rental housing have never been so great". The booklet, avail-

Welding the Piping for a Coolant System

By R. H. Harvey, Maintenance Engineer
Thompson Products, Inc.
Cleveland, Ohio

CENTRAL supply systems for machining coolant, used in only a few of the biggest industrial plants prior to the war, are now being incorporated in many large factories. The systems at Thompson Products, Inc., Cleveland, involving 100,000 feet of piping, are among the most modern in the country and are now being enlarged.

In a central coolant system, instead of each machine having its individual coolant tank and pump, an extensive piping network conveys coolant from central storage, filtering and pumping facilities to machines throughout the factory, and return.

The Thompson Products piping is practically all of arc welded construction to minimize flow resistance, and to minimize construction and maintenance costs.

The largest coolant piping system at Thompson begins with a 90,000-gallon tank in a separate building. In one section of the tank, filters screen out the sludge, which collects at the bottom of the tank and is removed by a continuous drag chain conveyor. The filtered coolant in the clean compartment is gravity-fed through 10-inch pipes to the main pumps in the oil basement of the factory. There are four pumps, each of 2,000-gallons-per-minute capacity. Only two pumps are operated at once, the other two being spares. 40-50 pounds pressure is maintained.

Fig. 4. Welding 45° elbow to 3-inch pipe.

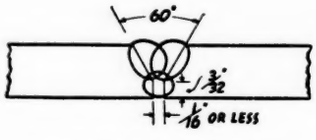


Fig. 2. Joint for 8-inch pipe.

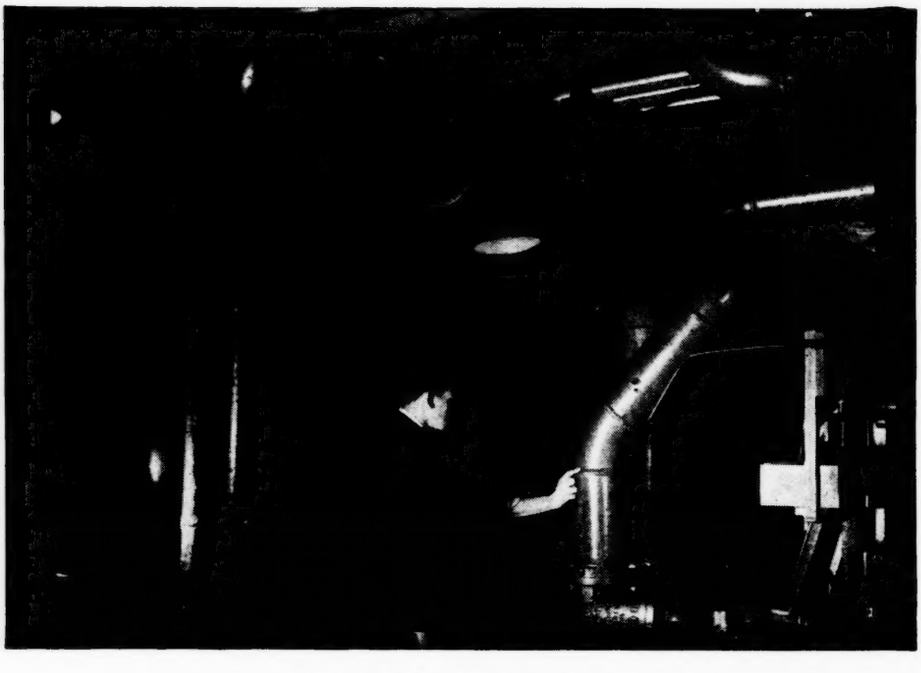


Fig. 1. Piping in oil basement.

Fig. 1 shows piping in the oil basement, including 8-inch piping which is the supply line from pumps to distribution piping in the plant. At left and top of picture are the 10-inch gravity-feed pipes to the pumps. The numerous bends were easily made by utilizing Tube Turns and arc welding.

Welding of the 8-inch pipe was done in three passes as sketched in Fig. 2. The 10-inch pipe was joined in four passes. Size 3/16" "Fleetweld 5" electrode and 150-170 amps. were used.

The supply and return pipes throughout the plant are 3 and 4-inch diameter. Supply piping is overhead. Return lines, now in the floor, are being converted to overhead to make rearrangement of machinery easier. Sump pumps—one for every four machines—will lift the sludge-laden coolant to the overhead return pipes, connecting from underneath at a 45-degree angle (see Fig. 3) to prevent clogging. The return pipes will be at a slight downhill angle so the sludge-laden coolant will flow by gravity back to the filtration tank.

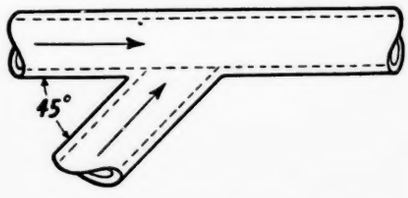


Fig. 3. Connection in return lines.

In Fig. 4 a 45-degree elbow is being welded onto a 20-ft. length of supply pipe, standard black steel of 3½ inches O.D. A 3-inch weld-neck flange has already been joined. Joints of 3 and 4-inch pipe are given a 75-degree bevel and welded in two passes with 1/8" "Fleetweld 5" electrode at 160 amps.

Complete procedures for welding all types of piping can be found in the "Procedure Handbook of Arc Welding Design and Practice", published by Lincoln Electric (price $1.50, postpaid). For a description of the book, write The Lincoln Electric Company, Dept. 481, Cleveland 1, Ohio.

Advertisement

able through the Federal Housing Administration, Washington 25, D. C., lists the many recent regulatory changes made to encourage rental projects. Included are summaries of relaxed provisions pertaining to rents, amortization, tax depreciation, working capital, mortgage review, property requirements, multiple mortgages, rehabilitation and conversion, site inspections and additional mortgage insurance.

Wagner-Ellender-Taft Again

Relayed reports of President Truman's views on the gulf between his housing policy and the attitude of what he calls "the real estate lobby", indicate that the Chief Executive is not at all satisfied with the way the housing program is going. Commenting on reports that only 45,000 housing "starts" were made during the first quarter of this year, as compared with 60,000 during the same period last year, he indicated that this record does not bear out industry's contention that building activity would increase with the removal of controls. (More drastic controls, including price ceilings on finished houses, were in effect during the first quarter of 1946). President Truman appeared particularly piqued at Congressional delay in passing a long-range housing bill embodying slum clearance and other provisions included in the Wagner-Ellender-Taft bill passed by the the Senate but shelved by the House during the last session of Congress.

That the bill probably has even less chance of being passed by the present Congress was indicated on April 22, when the Senate Banking and Currency Committee, by a vote of only 7 to 6, approved the bill for consideration by the Senate. The bill, designed to encourage construction of 15,000,000 homes by 1958, would cost the Federal Government approximately $7½ billion over a 45-year period, it is estimated by Senator Buck (R), Delaware, who opposed it.

Other Recommendations

How bitterly the bill is opposed by many influential members of the construction industry was revealed during the Senate Committee hearings. Herbert U. Nelson, executive vice president of the association testified that the housing problem could be quickly solved by removal of controls on the real estate industry. Among other recommendations he urged:

(1) That the National Housing Agency be abolished.

(2) That the Federal Housing Administration be returned to its previous status under a separate administrator—"that of a mortgage insurance agency."

(3) That Federal Home Loan Bank be returned to its original status, operating as a separate entity.

(4) That the Federal Savings Loan Insurance Corp., be re-established as a separate entity.

(5) That funds available under the Veterans Emergency Housing Act not expended or committed be returned to the Treasury. "We are advised that will approximate $335 million," he said.

Last but not least, he urged that a "15% across-the-board increase in rents be granted, and that all controls, including rent controls, be removed from new residential construction.

Despite such determined opposition, best guess in Washington is that rent controls will be continued in substantially their present form, with no blanket increase authorized.

Attacks on the credit provisions of the bill were made by Abner H. Ferguson of the United States Savings and Loan League and Douglas Whitlock of the Building Products Institute.

Ferguson said the credit provisions would "violate all the principles of sound lending as we know them." They would give, he added, "the most liberal credit terms to those whose financial condition makes them least eligible for credit."

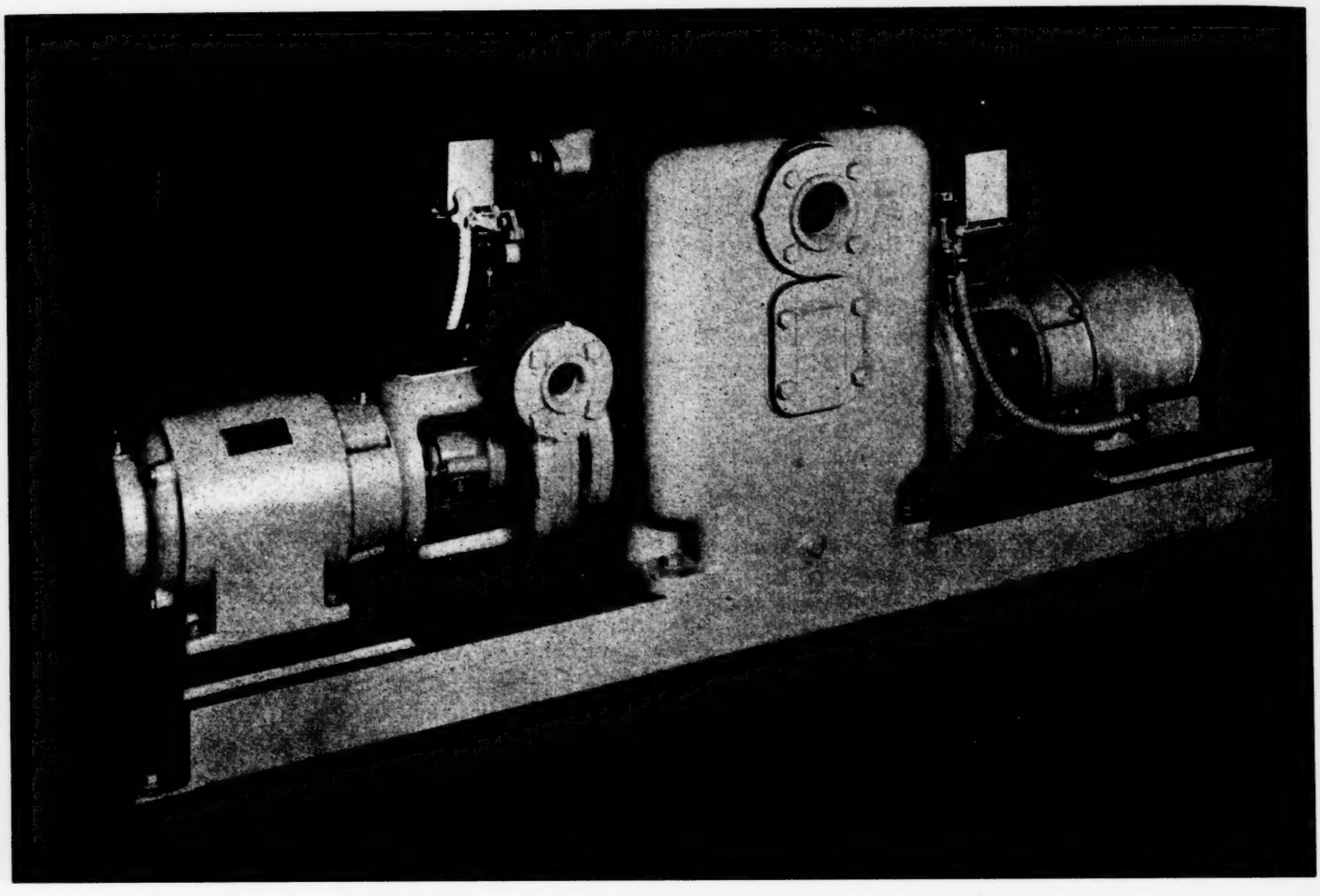
Reflecting much of industry's desire for freedom to build as it chooses are several amendments to the Veterans Emergency Housing Act, proposed by Representative Jesse Wolcott and approved by the House Banking and Currency Committee. This bill would, as suggested by realtors, abolish the Office of the Housing Expediter, together with all powers designed to expedite construction of the type of housing the Administration believes veterans need. The Administration takes the position that a few construction materials and incidentals are still in critically short supply, and that unless they are channeled into veterans' housing projects, there will not be enough to go around. The chance of veterans competing in such a market would be slight, Government spokesmen contend.

Hence, opposition to the Wolcott amendments is voiced by the American Legion, the American Veterans Committee, AMVETS, Disabled War Veterans and the Veterans of Foreign Wars. They are unanimous in their insistence that limitations on non-residential construction, limitations on the type of housing to be built, and allocation of scarce basic materials are essential to an effective veterans' program. The outcome of the battle between the Vets and the so-called "real estate lobby" will be interesting to watch. If the usual "who swings the most votes" pattern is followed, it would seem safe to predict that the Vets will come out on top.

Meanwhile, for what it may be worth, the construction volume total on which depends the distribution of heating, ventilating and air-conditioning equipment shows signs of dropping some 20 to 25 percent below the former official estimate of 15 billion dollars for new construction in 1947. Industry blames the drop on Government controls and high labor costs. Government blames the situation on high prices and big profits. Industry says it hasn't been given a chance to show what it could do. Government says that when price ceilings were removed, industry priced itself out of the market.

Doubtless the present hesitancy in the construction field will turn out to be due to a combination of circumstances, but heeding the "orange light" on the price matter may prove the starting point toward solution of a problem that is bitterly controversial. Increased production by labor at reasonable hourly rates "will build the stockpiles that will lower the cost of materials that will complete the job of building the house that Jack wants—at a price Jack can afford."

MAXIMUM EFFICIENCY FOR STEAM HEATING SYSTEMS ON HEATING, DRYING AND PROCESS WORK.



JENNINGS CONDENSATION PUMPS

Designed for steam systems on heating, drying, or process work, these pumps offer the most reliable and efficient means of returning hot condensate to boiler or hot well. In many cases these pumps will save costly installation of boilers in a pit to provide gravity flow of returns from basement radiators and heating devices.

The Jennings Condensation Pumps are sturdy and compact in construction, and combine receiving tank, pump and driving motor in a single assembly. Pumps are bronze fitted throughout, with tobin bronze shaft. The pumping impeller is of special design, made possible by our wide experience with return line heating pumps, and is especially adapted for handling hot water with the greatest possible efficiency.

Full automatic control is furnished by means of a ball float and float switch mechanism mounted on the receiving tank. All wiring is made up at the factory, and it is necessary only to connect the pump to the system and hook up the leads. Complete information regarding these economical pumps sent promptly on request.

THE NASH ENGINEERING COMPANY

SOUTH NORWALK, CONNECTICUT, U. S. A.

Refrigeration for Textile Mill Air Conditioning

P. L. DAVIDSON
Consulting Engineer, Philadelphia, Pa.

This study of the best methods of humidifying and cooling textile mills is based on an investigation of humidifying and cooling requirements for a group of textile mills. The article covers detailed analysis of refrigeration requirements for two comparative systems.

IN applying refrigeration to textile mill air conditioning, we do not attempt to produce the temperatures associated with comfort cooling. However, we do succeed in assuring the continued maintenance of conditions which afford the best textile processing and sustained worker efficiency—all at a cost, both for purchase and operation, that is frequently less than that of conventional systems which provide less satisfactory mill conditions. The fundamental problem in all applications of air conditioning in textile plants is the control of heat and humidity.

Prior to the period when proper air conditioning was introduced, it was necessary to locate textile plants in areas endowed with high humidity. Such areas required little or no artificial humidification. With the development of properly regulated humidification systems in textile mills, geographical location became less important. At present it is possible to operate satisfactorily even in areas of low natural humidity.

Before discussing the methods of humidifying and cooling textile mills, we should first consider the factors elemental in the determination of the inside temperature and humidity conditions to be maintained. Relative humidity, which, in textile operations, varies from 50 to 90%, fixed by the type of fiber being processed and the particular processes it is undergoing. In most cases, satisfactory working conditions for any particular fiber in process will obtain within a humidity range of 5% to 10%, provided that whatever humidity is selected is maintained with a variation of plus or minus 2%, and the process is adjusted to the selected humidity condition. In other words, maintenance of

constant humidity is more important than the set level of relative humidity.

Dry bulb temperature may also affect the process. Size formulas are somewhat affected by operating temperature as are pliability and other physical characteristics of certain synthetic fibers now being widely used. Here again, constancy of temperature is more important than the absolute level maintained. Operation of modern high speed looms, knitting and tricot machines can be very adversely affected by changes in temperature. All these machines, as well as size formulas and even to a certain extent fibers, can be adjusted to perform satisfactorily over a wide range of temperatures, providing that the selected temperature level is closely maintained during operation.

Spinning room of a southern textile mill. Air distribution is through pan type outlets. Atomized moisture for supplementary humidification is through atomizing nozzles uniformly spaced throughout the area. Bricked-up windows at right show location of relief dampers made of polished aluminum and which show up as bright spots in photograph.

Since relative humidity is determined by the process and fiber used, there is every reason to select the design temperature for its effect on the comfort and efficiency of the textile workers. Comfort depends on relative humidity as well as temperature, so since the humidity is fixed by the process, the dry bulb temperature for the same comfort condition must vary for each humidity.

To eliminate the necessity of discussing the different dry bulb temperatures for each humidity used in textile processes, we will talk of comfort in terms of effective temperature* which combines in one index the effect of relative humidity and temperature.

The index of effective temperature has been established through research and is generally accepted and well understood by heating and ventilating engineers. A tabulation of the various combinations of temperature and humidity which give an effective temperature of 75, 80, 85 and 90 follows:

Table 1—Combinations of Temperature and Humidity for Given Effective Temperature Air Velocity, 15-25 FPM

75 degrees ET			80 degrees ET			85 degrees ET			90 degrees ET		
D.B., F	W.B., F	R.H., %	D.B., F	W.B., F	R.H., %	D.B., F	W.B., F	R.H., %	D.B., F	W.B., F	R.H., %
77	73½	85	82.1	78.4	85	87.6	83.7	85	93.0	89.0	85
77½	72¾	80	82.8	77.8	80	88.5	83.0	80	94.2	88.5	80
78	72	75	83.6	77.2	75	89.5	82.5	75	95.5	88.0	75
78¾	71¼	70	84.4	76.6	70	90.5	81.9	70	96.6	87.7	70
79½	70½	65	85.3	75.8	65	91.6	81.3	65	98.0	87.1	65
80	69¾	60	86.2	75.0	60	92.7	80.6	60	99.4	86.6	60
81	68¾	55	87.2	74.2	55	93.9	79.8	55	101.0	86.0	55

*A subjective scale developed by the American Society of Heating and Ventilating Engineers, which takes into account humidity, temperature and air motion and which can be used to determine numerically those combinations of the three variables that give the same feeling of comfort is called the effective temperature scale.

A thorough study of data based on investigations warrants the statement that effective temperature in a textile mill should not exceed 80 with an 8-hour exposure for maximum efficiency of the workers. Temperatures above 85 deg ET (effective temperature), while not injurious to the health of the average worker in good physical condition, may bring about many undesirable physiological reactions to those whose resistance is lowered.

The maximum of 80 deg ET should probably be modified for northern New England and Canada or other areas where the maximum design outside wet bulb is 72F or less. As shown in Table 1, 80 deg ET means a dry bulb of 86.2F and a wet bulb of 75F at 60% R.H. and 15 to 25 fpm air velocity. This would not be considered a comfortable working condition for the worker who is living in areas such as Maine where the maximum outside dry bulb is 90F and the wet bulb rarely exceeds 72F. At 80 deg ET the mill temperature would be considerably higher than the outside conditions practically all of the time. For operations in Maine or Canada or similar climatic zones, it is our opinion that 75 to 77 deg ET should not be exceeded. The cost of maintaining an effective temperature of 75 to 77 in a mill in Maine would not be any more than the cost of maintaining an effective temperature of 80 in the Piedmont section of the Carolinas.

There are five different methods in common use today for humidification of textile mills:

(1) Use of atomizers to introduce moisture directly to the humidified space with dependence upon natural ventilation for air change and temperature control.

(2) Use of atomizers, with fan type air changing units, located in the windows introducing dry outside air, either directly into the space or through distribution ducts.

(3) Units to introduce air which carries in suspen-

sion droplets of unvaporized water in excess of the amount required to saturate the air at entering conditions. This air is properly referred to as "supersaturated."

(4) Humidity maintained by the introduction of dry saturated air (i.e., all water in vapor form) from an air washing chamber to the space to be conditioned.

(5) Desired room humidity achieved by introduction of a reduced quantity of dry saturated air with supplementation of humidity by direct atomization of moisture within the space.

Well water or water cooled by mechanical means can be employed readily for artificial cooling with either of the last two methods of humidification described.

For clarification, "units" is defined as standardized apparatus, usually of relatively limited capacity, used in multiple to handle a given application. "Central stations" are built-up systems designed and assembled specifically for the application in question and therefore unlimited in capacity; usually only one is employed to care for a given area or operation.

Without the use of a coolant such as well water, or mechanical refrigeration at any given outside wet bulb temperature and for the same inside relative humidity, equal average inside temperatures will be held by any of these five systems providing that they introduce the same quantity of outside air.

It is problematical that the simple atomized system will ever achieve this theoretical condition for it is solely dependent upon natural ventilation for the control of temperature and is therefore subject to the vagaries of the wind and manual control of window opening. Furthermore, it is obviously impossible to maintain the same temperature and relative humidity in the center of the room as near the windows.

With the system employing atomizers and fan type air changing units, the supply of outside air in sufficient quantity to afford maximum practical cooling can be positively assured and continuously maintained. However, the introduction of large quantities of outside air at isolated points along the walls makes impossible the maintenance of relative humidity within the desired tolerance. The addition of distribution ducts with properly spaced outlets to such units will improve the regulation of relative humidity.

An interesting point about the unit system, using dry outside air and atomizers, is that it is possible, and practical under many circumstances, to bring in more air than can be introduced from systems using saturated air. Under favorable conditions, it is therefore possible to maintain lower mill conditions with this system than with conventional central station, air washing systems. For practical reasons this condition does not apply where the maintained relative humidity is above 65%; or where the air washing system is designed to only partially saturate the air. In the latter case, air delivery from it may be increased to match the air capacity of the dry unit system with resulting equal room conditions.

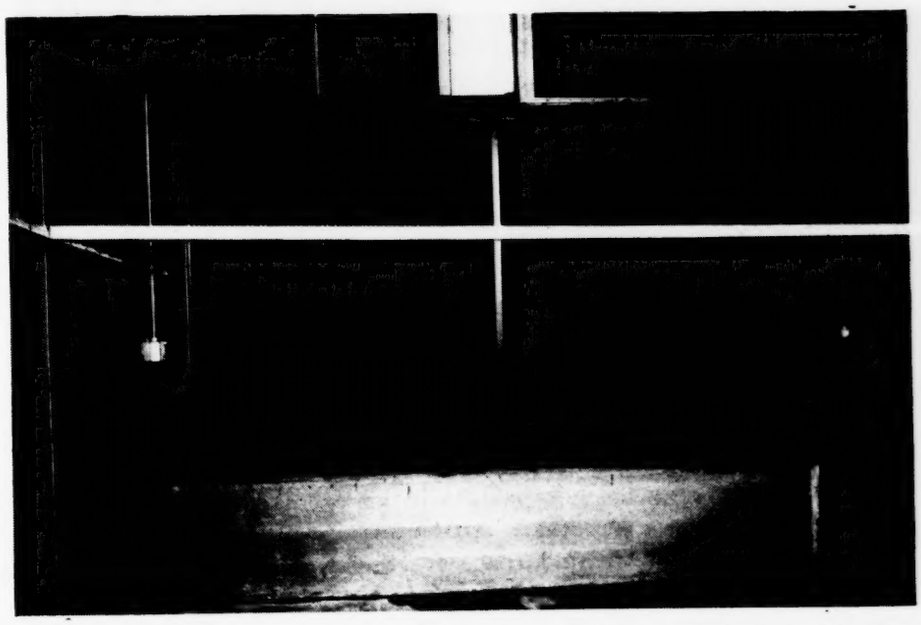
Units or central stations depending solely upon the introduction of saturated air from washing chambers and with properly designed duct systems, are capable of maintaining relative humidity with a variation of plus or minus 2%.

Central stations or units introducing reduced quantities of saturated air properly distributed and with supplementary humidification from atomizers in the room will also maintain relative humidity with a variation of plus or minus 2% throughout the room, but the reduced air quantity results in higher internal temperatures. Obviously the addition of sufficient quantity of a coolant at the proper temperature to either of these latter systems can result in the maintenance of any desired temperature in the room.

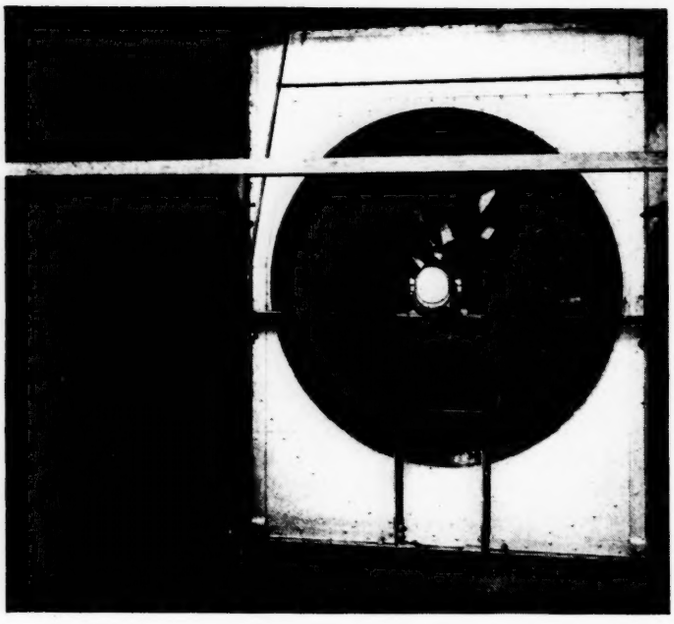
In practice, the following inside conditions can be anticipated with the various systems as outlined using outside air only, with no artificial cooling, and assuming an outside wet bulb temperature of 78F:

Table 2 — Temperatures for Various Operating Conditions

	75% R.H. in Mill		60% R.H. in Mill	
	D.B.	ET	D.B.	ET
Atomizers	90.5	85.8	96	87.5
Window fans and atomizer	86.7	82.6	92.6	85
Saturated air, central stations or units	86.7	82.6	93.8	85.8
Saturated air, central stations or units with supplemental atomizers in the room (no refrigeration)	89.5	85.0	97.8	88.8
Above with refrigeration for 80 deg ET	83.7	80.0	86.2	80.0



Discharge end of spray type humidifier as installed in a southern textile mill. Stainless steel eliminator plates remove droplets of free moisture from spray-saturated air. Reheaters are located above the humidifier which permits recirculation of air not treated in the humidifier chamber.



Inside humidifier discharge chamber with inlet of centrifugal blower which circulates air through system.

With an outside wet bulb temperature of 76F, and 80% R.H. maintained in the conditioned space, a system employing only saturated air will afford an internal effective temperature of 80 without the use of an artificial coolant. With this system, and at an outside wet bulb temperature of 78F, the room humidity would have to be raised to 90% R.H. to achieve an internal effective temperature of 80F. Therefore, with a maximum design wet bulb temperature of 76F, refrigeration will be required for the maintenance of 80 deg ET at any relative humidity below 80%; and similarly, at 78F W.B. refrigeration will be required for any relative humidity below 90% if 80 deg ET is to be achieved. Raising the limit of internal effective temperature will naturally increase the range of performance of systems without refrigeration. For example at 85 deg ET, refrigeration will only be required for maintenance of relative humidities below 67% at 78F W.B. outside; or for humidities below 57% at 76F W.B.—but these higher effective temperatures are considered undesirable.

With any system, the basic amount of air required increases rapidly as the maintained level of relative humidity is increased, or as the heat load within the space increases. Where dependence is solely upon outside air for cooling, with the consequent necessity of always reaching for the closest approach to outside wet bulb temperature, air quantity can assume troublesome proportions at high humidities and moderate loads, or at lower humidities with high density loads.

In weave rooms with modern high speed looms any relative humidity above 75% results in an excessive amount of air. In many cases with central station systems it is difficult to find space for ducts or for equipment housing. With the unit system, the number of units required becomes excessive.

The modern twister operation, using double-deck twisters, may have a heat load as high as 200 Btu per sq ft. Here, again, though the maintained relative humidity may be only 50 to 60%, the amount of air required is excessive. It has, therefore, been common practice with high relative humidity or high heat load operations to compromise and to use a central station or unit system of reduced capacity with supplementary atomization in the room or units of the type delivering super saturated air, both of which methods are thermo-dynamically equivalent.

The air capacity requirement of systems is an important consideration. Coupled with the desirability of maintaining reasonable effective temperatures, it forms the basis for the economic application of refrigeration of textile mill air conditioning. Since refrigeration makes practical a substantial reduction in the required air delivery of systems, a comparative analysis of the system depending solely upon the use of saturated air with that employing saturated air in combination with supplementary atomization, should be interesting.

For analysis purposes, we have assumed an outside wet bulb temperature of 76F. We also use the term "heat content" to describe the sigma function of air, i.e., sensible heat in the air above 0F plus the latent heat in the water vapor at saturation, but not including the heat of the liquid.

In practical operation, air would leave the humidifying washer at about 77F D.B., 76F W.B., 96% R.H. and a heat content of 38.7 Btu per lb. For purposes of this analysis, we will assume that our problem is one of absorbing a heat load of 1,000 Btu per min or 60,000 Btu per hr.

For case *A*, we will assume a system in which the saturated air is introduced in sufficient quantity to rise in dry bulb temperature to a point resulting in a 60% R.H. while absorbing the heat load and without the further addition of moisture within the space. The final condition achieving this result will be 91.6F D.B., 79.7F W.B., and, at this condition, the heat content of the air will be 42.2 Btu per lb. Each pound of air will have absorbed the heat quantity equal to the heat content rise from its entering condition to its final condition, or 3.5 Btu. The pounds of air required to absorb the load of 1,000 Btu per min will be approximately 286 lb per min. Assuming an air density of 13.5 cu ft per lb (which we will use throughout this analysis) the required air quantity with system *A* will be 3,860 cfm.

For system *B*, we will assume a system in which a minimum practical quantity of saturated air is delivered to the space for heat absorption, and in which additional moisture is introduced to the space by atomization and evaporation to maintain the desired relative humidity. The air will be allowed to rise to a final condition of 83.5F W. B. At 60% R.H., this will result in a final mill condition of 95.8F D.B., 83.5F W.B. and the heat content of the air will be 46.2 Btu per lb. To achieve this final condition, it will be necessary to add approximately 20 grains of moisture per pound of air by direct atomization. With this system, the required air quantity to absorb the heat load of 1,000 Btu per min will be approximately 133.2 lb per min based on the heat content rise of 7.5 Btu per lb. Assuming an air density of 13.5 cu ft per lb, the

required air quantity will be approximately 1,800 cfm.

While system *B* requires far less air than system *A*, this reduction in air quantity is achieved by absorbing more heat per pound of air delivered and, therefore, the final mill temperature for system *B* will be above that for system *A*. With the entering condition attained by evaporative cooling from 76F W.B., the resulting effective temperature in the space will be for system *A*, 84.1 deg ET, for system *B*, 87.2 deg ET. The air quantity required for system *B* is, however, less than one-half that required by system *A*.

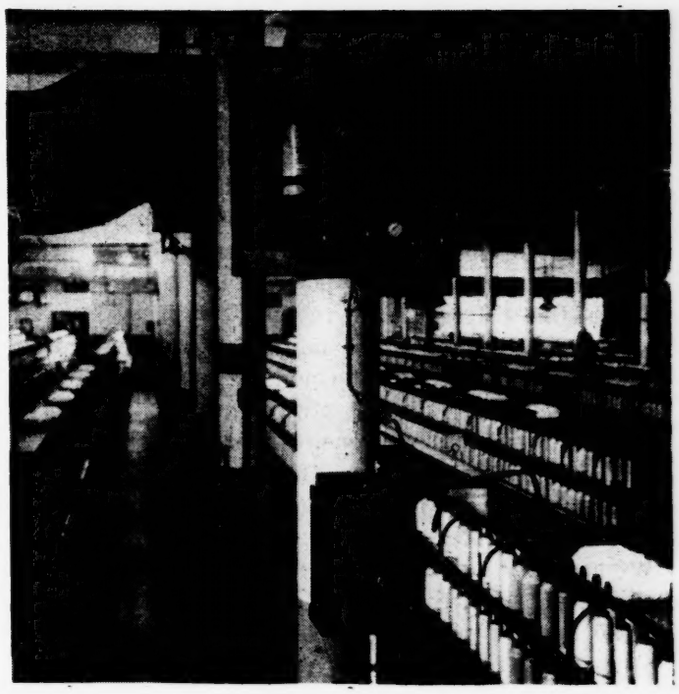
To carry the analysis a little further, let us assume that it is desired to hold with either system *A* or *B*, a final effective temperature within the mill not exceeding 80 deg ET at 60% R.H. To achieve this result, either system would have to be designed to maintain a final mill condition of 86.2F D.B. and 75.0F W.B. at which condition the heat content of the air would be 37.7 Btu per lb.

Reversing the selection procedure used in the previous cases, i.e., assuming the final mill condition to be constant and selecting an entering air condition necessary to achieve this, we find that with system *A*, it would be necessary to introduce air from the humidifying washer at 72.0F D.B., 71.0F W.B., 96% R.H., 70.5F dew point, at which condition the air has a heat content of 34.2 Btu per lb. The heat content rise from entering condition to mill condition is 3.5 Btu per lb, as in previous case *A*, so that the required air quantity will remain at 3,860 cfm.

Following the procedure for system *B*, and to achieve the final mill condition of 80 deg ET, it would be necessary to introduce air from the humidifying washer at 66.9F D.B., 65.9F W.B., 96% R.H., 65.4F dew point. Heat content of the air will be 30.2 Btu per lb. To achieve this condition supplementation with approximately 19 grains of moisture per pound of air will be required. Under case *B* revised, the heat content rise remains at 7.5 Btu per lb of air so that the air quantity required to absorb the heat load will remain at 1,800 cfm.

Either of these systems will maintain a room condition of 80 deg ET if air is delivered at the required entering condition. System *A* will accomplish this result without artificial cooling when the outside wet bulb temperature drops to 71.0F and system *B* will similarly meet the condition when the outside wet bulb temperature drops to 65.9F. In either case, the condition can be met by use of some outside coolant to produce the desired delivery temperature from the humidifying air washer during those periods when the outside wet bulb is above the desired level. Coolant of proper quantity and temperature level can be obtained from any source. Sometimes wells are successfully used, but more often refrigeration is employed.

In applying a coolant to a humidifying washer, it is necessary to assume some temperature differential between the leaving air temperature and the chilled water temperature within the washer pan after it has done its work. This is generally accepted as 1.0F. Therefore, for system *A* to maintain 80 deg ET, we would assume a final water temperature of 71.0F and for system *B*, a temperature of 65.9F. There are also



Master hygrostat control station. Above the control box is the master control panel that holds the air filter and pressure reducing station and the various master control relays.

practical limits to the amount of rise that can be obtained in passing coolant through the washer which is generally accepted at 7.0F. For system *A*, water would be delivered at 64F and for system *B* water would be delivered at 58.9F. In either case, the quantity of cooling effect or refrigeration required is identical as can be seen when it is considered that in each case the stipulated air quantity absorbs 1,000 Btu per min in rising from its entering condition to mill condition and that conversely the removal of 1,000 Btu per min will be required to reduce it from mill condition to entering condition. The only difference between the requirements of systems *A* and *B* lies in the temperature level at which the coolant must be introduced to the humidifying air washer.

As demonstrated, the refrigeration load for systems *A* and *B* is identical. Significance of the difference in temperature level can be roughly analyzed by considering the approximate temperature heads involved in a refrigeration system. It is necessary to have a temperature differential between the evaporating refrigerant in a water cooler and the chilled water. Practical considerations generally dictate that this be approximately 10.0F below the leaving or low water temperature. Similarily, it is necessary to have a temperature differential between the refrigerant vapor in the condenser and the condensing medium which we will assume to be water from a cooling tower. This also must be approximately 10.0F and can be established for both *A* and *B* at 105.0F, assuming that the cooling tower will cool the condensing water from 95.0F to 85.0F versus 76.0F W.B.

The power required for refrigeration will be roughly proportional to the temperature differential between the suction temperature and the condensing temper-

ature. In accordance with these considerations, the required suction temperature for system *A* will be approximately 54.0F and the temperature difference beween this and the condensing temperature will be 51.0F. For system *B*, the evaporating temperature will be 48.9F and the temperature difference between this and the condensing temperature will be 56.1F.

The ratio of *A*/*B* will be approximately 91%. This indicates that slightly less power will be required to produce the refrigeration required with system *A*, than with system *B*, and might also indicate that slightly smaller equipment could be used. These differences, however, are not always practically achievable for a number of reasons. Primarily it is because refrigeration machines are finite in their design so that it is generally impossible to get the full benefit of such slight differences in conditions.

We can thus see that there is little important practical difference in the refrigeration required to maintain 80 deg ET with system *A* as compared with *B*. System *B*, at the same time, however, uses less than one-half the air required for *A*.

Importance of this consideration can be seen by quickly analyzing two cases. For case *A*, we will assume a typical high density load as encountered with double-deck twisters. In certain cases, the load density in this type of application will run as high as 200 Btu per hr per sq ft of floor space. Returning to the base load of 1,000 Btu per min or 60,000 Btu per hr, the air quantities established would care for approximately 300 sq ft which, assuming a 15 ft ceiling height, would be 4,500 cu ft. Using system *A*, 3,860 cfm would be required which represents an air change within the space once every 1.165 min or 52 changes per hour. We believe that air change at this rate will generally be conceded impractical both from the standpoint of system size and the very physical difficulty of handling this quantity of air through a manufacturing area. Using system *B* to care for the same load within the same space, only 1,800 cfm would be required which would represent an air change once every 2.5 min or 24 changes per hour. This is a high rate of air change, but is within practical limits.

It may not be fair to assume a load of such great density, so we will repeat the analysis for an area with a loading of 100 Btu per hr per sq ft which is a moderate load in typical spinning room practice. In this case, our unit of capacity would handle 600 sq ft or 9,000 cu ft. Our air change with system *A* would be once every 2.33 min or 24 changes per hour; with system *B*, once every 5 min or 12 changes per hour.

In either case, the advantages of handling approximately one-half the air quite obviously favor system *B* and in most cases the actual saving in first cost of air systems will more than offset the entire cost of refrigeration required for use in system *B*. We wish to reemphasize the point that the cost of system *A* without refrigeration is generally comparable to the cost of system *B* with refrigeration.

The comparison of operating cost is more involved since it requires study not only of average weather conditions but also of power cost including both demand and consumption considerations. However, since air circulation must be continuous throughout the entire operating year and since system *B* circulates approximately one-half the air required by system *A*, it can be stated that the resulting power saving will substantially offset the cost of power required for refrigeration and air compressor operation. Let us emphasize that this statement also is made relative to system *B* with refrigeration compared with system *A* without refrigeration.

There is little object in comparing the two systems when both are used with refrigeration. System *B*, with refrigeration, will do everything that system *A* will, and at a substantial saving in first cost and operating cost for air circulation systems. It is true that the refrigeration equipment used with system *A* will not have to operate as many hours during the season but it will cost approximately as much to purchase as that required with system *B*, and it will require approximately the same peak demand for electrical energy which is usually the predominating factor in determining the power cost of refrigeration operation.

System *B* offers one major operating advantage. With system *A*, the quantity of air delivered to the space must be accurately regulated to the load requirement if the desired relative humidity is to be continuously maintained. This presents a number of practical difficulties as recirculating controls are quite costly and volume control cannot give satisfactory results. With system *B*, the final relative humidity is controlled directly within the space or zone by starting and stopping the atomizers. Under any reasonable load condition, there is a deficiency of humidity in the delivered air. Changes in load will result in some variation in temperature, but humidity can be controlled accurately at all times within a multiplicity of zones on a single system.

This analysis definitely proves the following:

(1) With an outside wet bulb of 76F, the central station or unit system employing saturated air with no supplementary evaporation can only maintain 80 deg ET when maintaining an inside relative humidity of 80% or higher.

(2) With an outside wet bulb of 78F, the inside relative humidity in the room must be 90% or higher if 80 deg ET is to be maintained. Lower room humidities than those must result in higher effective temperatures if this system is employed without refrigeration. By reducing the size of the central station equipment and introducing supplementary evaporation, the saving in initial cost, operation, and maintenance of air handling equipment will substantially offset the cost of the necessary refrigeration machinery. This results in benefits of reduced effective temperature in the conditioned space at a minimum expense.

When it is considered that a central station system with reduced air circulation capacity, supplementary evaporation and refrigeration can be designed to produce any desired effective temperature, and can be installed more easily and at a cost no greater than a full evaporative central station system without refrigeration, there seems little doubt that we will see a vast increase of the use of refrigeration in conjunction with textile mill air conditioning.

The Coefficient of Performance of the Heat Pump

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A clear understanding of the coefficient of performance of the heat pump is approached through an analysis of refrigeration cycles and fundamental thermodynamics.

WITH the present accelerated interest in the heat pump, it is fitting to inquire into its operating characteristics; especially, its coefficient of performance, which, after all, is the primary source of interest. If the coefficient of performance is high enough to bring electricity economically into the competitive field of house heating, all items of expense considered, then the heat pump should be promoted with every means available. If the coefficient of performance is too low to accomplish this, then too much time and money is being wasted on a mechanical toy.

With this in mind, let's start with fundamentals, make a review of refrigeration cycles, and attempt to obtain a better understanding, both qualitatively and quantitatively, of the so-called coefficient of performance.

Now the coefficient of performance of a machine may be defined as the desired effect divided by the energy supplied to produce that effect. Specifically, with respect to the heat pump as a heating device, the coefficient of performance will be the heat removed in the condenser divided by the heat equivalent of the energy supplied to the system.

Several diagrams have been devised for studying the operation of refrigeration machines and refrigeration cycles. First among these is the P-V diagram or indicator card, illustrated in Fig. 1. Using absolute pressure, P, as ordinates and volume, V, as abscissae,

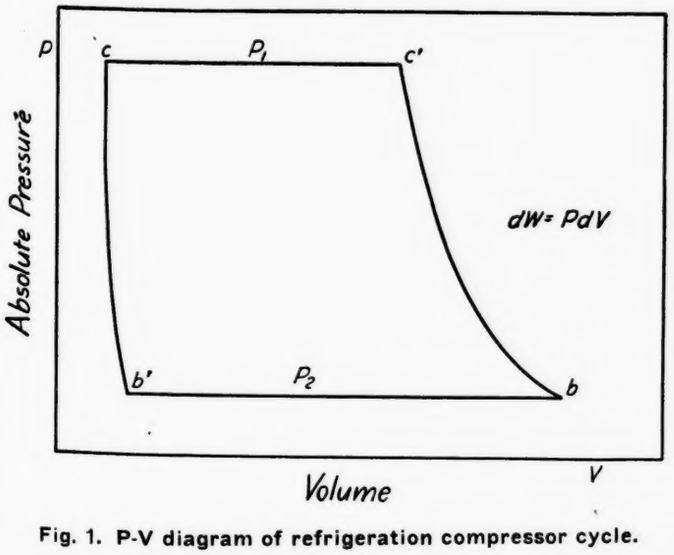

Fig. 1. P-V diagram of refrigeration compressor cycle.

the diagram describes the operation of a refrigeration compressor, as the vapor is compressed adiabatically from state-point b to state-point c'; discharged at constant pressure from state-point c' to state-point c; as clearance vapor re-expands adiabatically from state-point c to state-point b'; and as a new charge of vapor is drawn into the compressor from state-point b' to state-point b. Increments of area on this chart represent increments of work, thus,

$$dW = PdV$$

Where dW is an increment of work, dV an increment of volume. The area bc'cb'b represents the net effective work done on the vapor. This diagram is helpful in studying the *compressor* operation, but a diagram is needed that will give a complete picture of the whole refrigeration cycle.

A chart on which absolute temperature, T, is the axis of ordinates and entropy, S, that of abscissae, provides a convenient means of following a pound of refrigerant from one state-point to another, throughout the entire cycle. Fig. 2 is a temperature-entropy, T-S, diagram. On it have been drawn two curved lines. The one passing through the point e, is the locus of all state-points of saturated liquid refrigerant. To the left of this line, is only sub-cooled liquid. The curved line passing through the point c, is the locus of all state-points of saturated vapor refrigerant. To the right of this line, is only super-heated vapor. Between these two lines, the refrigerant is a mixture, of varying quality. On this chart, increments of area represent increments of total heat or enthalpy, thus,

$$dH = TdS$$

Where dH is an increment of total heat, dS an increment of entropy, and the total heat (above some reference point, such as g) for any state-point such as e, is represented by the area $o\ g\ e\ j$; or, for the state-point c, by the area $o\ g\ e\ c\ k$.

The ideal Carnot cycle consists of two isotherms between two isentropics; that is, two constant-temperature processes and two constant-entropy—or reversible adiabatic—processes. Operation of the two isentropic processes is between the temperatures of energy supply, T_2, and energy discard, T_1. Starting with a pound of refrigerant at the state-point b, having a total heat represented by the area $o\ g\ m\ b\ k$, let it be compressed, adiabatically, to the state-point c, having a total heat of $o\ g\ e\ c\ k$. The difference between these two areas—that is, $m\ b\ c\ e\ m$—then represents the heat equivalent of the work of compression. At constant temperature, condense the vapor refrigerant at c, to the state-point e, removing or discarding heat represented by $j\ e\ c\ k$.

Now expand, adiabatically, the liquid refrigerant at *e*, to the state-point *a*. In doing so, the refrigerant will perform an amount of work whose thermal equivalent is *m a e m*. Thus, the net work of the cycle is *m b c e m* less *m a e m*, or *a b c e a*; and the coefficient of performance is:

$$\text{C.O.P.} = \frac{\text{j e c k}}{\text{a b c e a}} = \frac{T_1(S_1\text{-}S_2)}{(T_1\text{-}T_2)(S_1\text{-}S_2)} = \frac{T_1}{T_1\text{-}T_2}$$

Table 1 shows values of this ideal coefficient of performance of the Carnot cycle, for various temperature combinations.

The theoretical cycle of the refrigerant proper, is shown in Fig. 3, on a T-S chart. It approaches, somewhat more closely, the practical refrigeration cycle and differs from the Carnot cycle, in two important respects:

(1) The refrigerant leaving the evaporator and entering the compressor, state-point *b*, is a saturated vapor, which becomes super-heated as it is compressed adiabatically to state-point *c*; and

Table 1—Coefficient of Performance of Carnot Cycle

(Highest Conceivable, But Actually Unattainable)

Saturated Condensing Temp., F	Saturated Suction Temperature, F							
	50	40	30	20	10	0	—10	—20
90	13.75	11.00	9.17	7.86	6.87	6.11	5.50	5.00
96	12.09	9.93	8.42	7.32	6.47	5.79	5.25	4.79
102	10.81	9.06	7.81	6.85	6.11	5.51	5.02	4.61
107	9.95	8.46	7.36	6.52	5.85	5.30	4.85	4.46
112	9.23	7.94	6.98	6.22	5.61	5.11	4.69	4.33
117	8.61	7.49	6.63	5.95	5.39	4.93	4.54	4.21

$$\text{C.O.P.} = \frac{Tc}{Tc\text{-}Ts} = \frac{tc + 460}{tc\text{-}ts}$$

(2) The expansion process is not adiabatic or isentropic. No energy is given up in performing work. No energy is given up at all; no work is performed. The refrigerant has the same total heat content at the end of the process, as it had before. The state-point *f* must be at such position as to make the area *o g m f n* equal to *o g e j*.

The first of these differences causes the so-called "super-heat horn" at *c* which, obviously, represents a loss of energy, as compared with the Carnot cycle. The total heat of the vapor before compression, is *o g m b k*, and after compression, *o g e d c k*. So the work of compression is *m b c d e m*. Since, in expanding, the refrigerant performs no work, the area *m b c d e m* represents the net energy input to the cycle. The energy rejected in the condenser is *o g e d c k* less *o g e j*, or *j e d c k*; and the coefficient of performance is:

$$\text{C.O.P.} = \frac{\text{j e d c k}}{\text{m b c d e m}} =$$

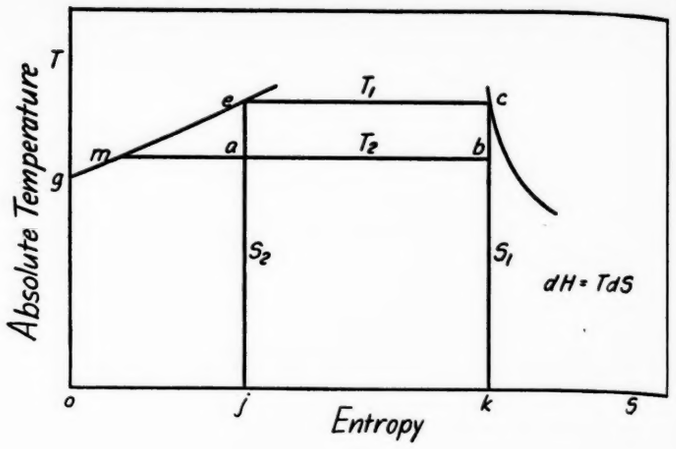

Fig. 2. Temperature-entropy diagram follows refrigerant from one state-point to another throughout the entire cycle. Left of curve through point *e* is liquid. Right of curve through point *c* is vapor.

$$\frac{\text{Total heat at c — Total heat at e}}{\text{Total heat at c — Total heat at b}}$$

Manifestly, it would be an endlessly complicated job to prepare, for the refrigeration cycle of Fig. 3, a table similar to Table 1 for the Carnot cycle of Fig. 2. A much easier procedure is available.

The Mollier diagram is drawn on a chart using absolute pressure, *P*, as the ordinates and total heat, *H*, as the abscissae, as shown in Fig. 4. The constant temperature processes of Fig. 3,—condensation from *d* to *e* and evaporation from *f* to *b*—are also constant pressure processes. They are shown as such in the Mollier diagram on the P-H chart. Here again two up-and-down curved lines are drawn; one representing a saturated liquid refrigerant and one a saturated vapor. Between these two lines, lies a mixture. Areas and increments of area on this chart, have no physical significance. However, calculation of coefficient of performance for any set of conditions, is enormously simplified.

Given a pound of refrigerant at state-point *b*, let it be compressed along a constant-entropy, or isentropic line, to state-point *c*, where it is a superheated vapor, as before in Fig. 3. Now this superheated vapor is cooled to state-point *d* and condensed to state-point *e*,

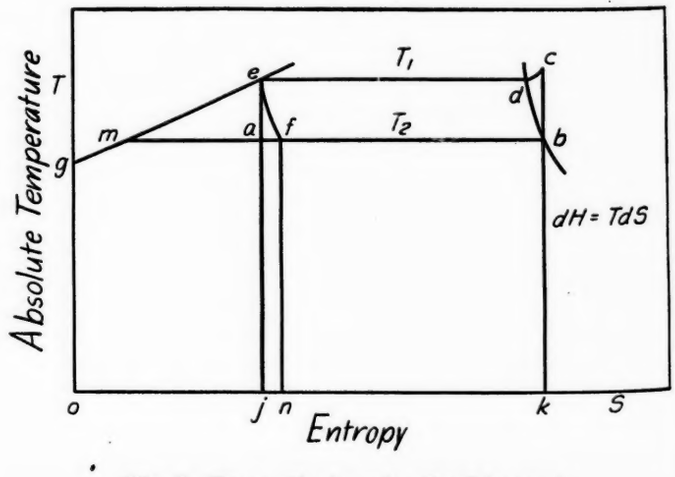

Fig. 3. Theoretical cycle of refrigerant.

Table 2 — Theoretical Coefficient of Performance of Freon-12 Refrigerant

Suction vapor and liquid refrigerant are assumed at saturation conditions.
Figures in lines 1, 2, and 3, represent Btu per pound.

Saturated Condensing Temp., F	Line Number	Item	Saturated Suction Temperature, F							
			50	40	30	20	10	0	—10	—20
	1	Ht. of Suction	83.8	82.7	81.6	80.5	79.4	78.2	77.0	75.9
	2	Ht. of Discharge	88.7	88.9	89.2	89.5	89.9	90.2	90.8	91.3
90	3	Ht. of Liquid	28.7	28.7	28.7	28.7	28.7	28.7	28.7	28.7
	4	C.O.P.	12.24	9.71	7.96	6.76	5.83	5.13	4.50	4.06
	1	Ht. of Suction	83.8	82.7	81.6	80.5	79.4	78.2	77.0	75.9
	2	Ht. of Discharge	89.3	89.6	89.9	90.2	90.6	90.9	91.3	92.0
96	3	Ht. of Liquid	30.2	30.2	30.2	30.2	30.2	30.2	30.2	30.2
	4	C.O.P.	10.75	8.61	7.19	6.19	5.39	4.78	4.27	3.84
	1	Ht. of Suction	83.8	82.7	81.6	80.5	79.4	78.2	77.0	75.9
	2	Ht. of Discharge	89.9	90.2	90.5	90.8	91.3	91.6	92.1	92.6
102	3	Ht. of Liquid	31.7	31.7	31.7	31.7	31.7	31.7	31.7	31.7
	4	C.O.P.	9.54	7.80	6.61	5.74	5.01	4.47	4.00	3.65
	1	Ht. of Suction	83.8	82.7	81.6	80.5	79.4	78.2	77.0	75.9
	2	Ht. of Discharge	90.5	90.8	91.0	91.4	91.8	92.2	92.7	93.3
107	3	Ht. of Liquid	32.9	32.9	32.9	32.9	32.9	32.9	32.9	32.9
	4	C.O.P.	8.60	7.15	6.18	5.37	4.75	4.24	3.81	3.47
	1	Ht. of Suction	83.8	82.7	81.6	80.5	79.4	78.2	77.0	75.9
	2	Ht. of Discharge	91.0	91.3	91.6	92.0	92.4	92.7	93.3	93.9
112	3	Ht. of Liquid	34.2	34.2	34.2	34.2	34.2	34.2	34.2	34.2
	4	C.O.P.	7.89	6.64	5.74	5.03	4.48	4.03	3.63	3.32
	1	Ht. of Suction	83.8	82.7	81.6	80.5	79.4	78.2	77.0	75.9
	2	Ht. of Discharge	91.6	91.8	92.1	92.5	92.9	93.2	93.7	94.4
117	3	Ht. of Liquid	35.4	35.4	35.4	35.4	35.4	35.4	35.4	35.4
	4	C.O.P.	7.21	6.20	5.40	4.76	4.26	3.85	3.49	3.19

$$\text{C.O.P.} = \frac{(\text{Ht. of Discharge}) - (\text{Ht. of Liquid})}{(\text{Ht. of Discharge}) - (\text{Ht. of Suction})}$$

For 10 degrees suction superheat and 10 degrees liquid subcooling, above values of C.O.P. may be increased about 4%.

all at constant pressure. At state-point *e*, the refrigerant enters the expansion valve, and emerges as a mixture at state-point *f*, without any change in its total heat content. At state-point *f*, the refrigerant enters the evaporator and is vaporized, at constant pressure, to state-point *b*. See, now, how simple becomes the calculation of coefficient of performance!

$$\text{C.O.P.} = \frac{\text{Total heat at c} - \text{Total heat at e}}{\text{Total heat at c} - \text{Total heat at b}}$$

Each of these factors is read directly on the horizontal axis of the chart. Now we may prepare Table 2, with ease. This has been done by using a Mollier diagram for Freon-12 prepared by Kinetic Chemicals, Inc.

Now the coefficients of performance shown in Table 2, are for Freon-12 refrigerant, proper. Given a compressor free of losses, driven by an electric motor free of losses; Freon entering the compressor as a saturated vapor and, the expansion valve, as a saturated liquid; the several units of the system being connected by frictionless copper tubing; then, the coefficients of performance of Table 2, would be attained.

It happens that, unlike ammonia, Freon-12 vapor, when heated, increases in refrigerating effect more rapidly than in volume. Thus, as would be expected, the coefficient of performance may be somewhat increased by super-heating the suction vapor. Likewise, an increase in the coefficient of performance results from sub-cooling the liquid in the condenser. Most expansion valves are set for 10 degrees suction superheat, and 10 degrees of liquid sub-cooling is not unusual. These two favorable conditions combined, result in an average increase in the coefficient of performance of about 4%. The dotted lines on Fig. 4 show the effect of vapor super-heating and liquid sub-cooling.

Line 3, in Table 4 shows coefficients of performance for Freon-12 with the combined effects of 10 degrees suction super-heat, 10 degrees liquid sub-cooling, 85% overall compressor efficiency, and 85% motor efficiency. Both of these efficiencies are well beyond the reach of present-day equipment of sizes up to 15 hp, certainly.

In Table 3, coefficients of performance for an actual commercial compressor, have been shown, as calculated from manufacturer's "Performance Ratings." Referring back to Fig. 3, it was shown earlier in this paper that

$$\text{C.O.P.} = \frac{\text{j e d c k}}{\text{m b c d e m}} =$$

$$\frac{\text{heat rejected in the condenser}}{\text{heat equivalent of work of compression}}$$

but since total heat at *f* is equal to the total heat at *e*,

area o g e j = area o g m f n

or o g m a j + m e a m = o g m a j + j a f n

therefore area m e a m = area j a f n

so area j e d c k = n f a e d c k + j a f n =
n f a e d c k + m e a m
and n f a e d c k + m e a m = m b c d e m + n f b k
Thus j e d c k = m b c d e m + n f b k

That is to say, the heat rejected in the condenser is equal to the heat equivalent of the work of compression plus the refrigerating effect. Actually, the denominator of the defining fraction some lines above, should read "energy input to system," instead of "heat equivalent of work of compression." But in the theoretical cycle of Fig. 3, where no losses are considered, the work of compression is the only energy input to system. In the actual commercial system:

$$\text{C.O.P.} = \frac{\text{heat equivalent of work of compression} + \text{refrigeration effect}}{\text{heat equivalent of motor input}}$$

In calculating the C.O.P. shown in Table 3, it was assumed that the heat equivalent of the work of compression is the same as the heat equivalent of the brake horsepower of the motor. This is not true; the condenser does not have to absorb or dissipate all of the energy delivered to the compressor shaft. Some 15% or more of this, constitutes mechanical losses—bearing friction, wire-drawing of the vapor through valve

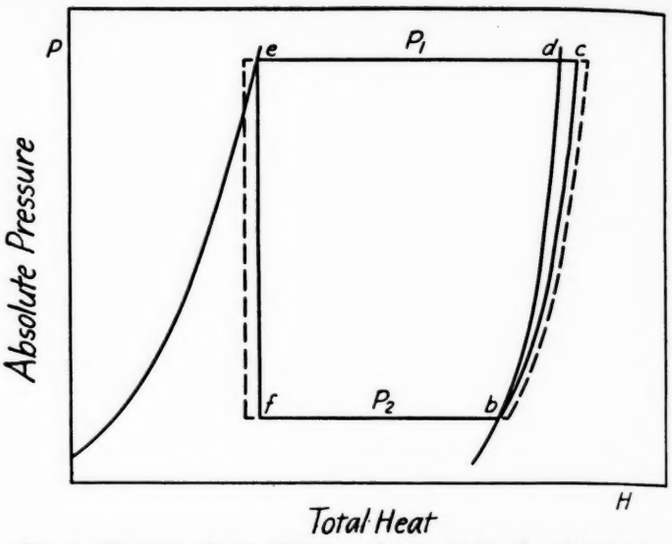

Fig. 4. Pressure-heat diagram shows state of refrigerant.

ports, etc. Obviously then, the coefficients of performance shown in Table 3, are liberal. For instance, the C.O.P. of 2.68 found in the example below Table 3, is at least 5% too high. However, a close study of compressor efficiencies, would not be appropriate here.

Table 4 is presented by way of summary. It is self-explanatory. The coefficient of performance of small

Table 3 — A Commercial Coefficient of Performance of Compressor Using Freon-12

Data in lines 1 and 2 taken from performance ratings of manufacturer.
All motors are 5 hp except where asterisk indicates 3 hp (see note below).

Saturated Condensing Temp., F	Line Number	Item	Saturated Suction Temperature, F							
			50	40	30	20	10	0	—10	—20
90	1	Ref. Eff., Mbh[1]	67.6	55.8	45.8	37.0	29.7	23.8	18.6	14.4
	2	Kw Input	3.48	3.55	3.52	3.40	3.22	3.00	2.95	2.72
	3	Brake H.P.	3.85	3.94	3.90	3.76	3.59	3.33	3.13*	2.94*
	4	C.O.P.	6.51	5.43	4.64	4.01	3.53	3.15	2.64	2.36
96	1	Ref. Eff., Mbh	65.5	54.0	44.3	35.8	28.8	22.9	17.9	14.0
	2	Kw Input	3.69	3.75	3.67	3.52	3.30	3.09	3.02	2.75
	3	Brake H.P.	4.10	4.18	4.10	3.90	3.65	3.40	3.23*	2.96*
	4	C.O.P.	6.03	5.05	4.37	3.80	3.38	2.99	2.53	2.29
102	1	Ref. Eff., Mbh	63.6	52.4	43.0	34.6	27.8	22.1	17.4	13.5
	2	Kw Input	3.92	3.91	3.82	3.65	3.42	3.17	3.08	2.80
	3	Brake H.P.	4.37	4.36	4.29	4.05	3.80	3.50	3.30*	3.00*
	4	C.O.P.	5.58	4.76	4.13	3.60	3.21	2.86	2.45	2.21
107	1	Ref. Eff., Mbh	61.7	50.9	41.7	33.6	26.9	21.5	16.8	12.9
	2	Kw Input	4.16	4.10	3.95	3.77	3.51	3.23	3.13	2.82
	3	Brake H.P.	4.61	4.52	4.40	4.17	3.90	3.60	3.35*	3.02*
	4	C.O.P.	5.17	4.46	3.92	3.43	3.07	2.78	2.37	2.14
112	1	Ref. Eff., Mbh	60.1	49.4	40.5	32.5	26.2	20.9	16.3	12.5
	2	Kw Input	4.38	4.28	4.11	3.89	3.60	3.30	3.17	2.85
	3	Brake H.P.	4.80	4.70	4.55	4.33	4.00	3.65	3.40*	3.05*
	4	C.O.P.	4.83	4.20	3.71	3.28	2.96	2.68	2.30	2.08
117	1	Ref. Eff., Mbh	58.4	48.0	39.4	31.7	25.4	20.2	15.7	12.1
	2	Kw Input	4.61	4.45	4.25	3.98	3.68	3.36	3.02	2.87
	3	Brake H.P.	5.10	4.93	4.70	4.42	4.10	3.70	3.32	3.07*
	4	C.O.P.	4.53	3.98	3.54	3.16	2.86	2.34	2.34	2.03

NOTE:—All motors are 220 Volt, 3 Phase, 60 Cycle, 1800 rpm, High-starting-torque, Low-starting-current, Induction motors.

$$\text{C.O.P.} = \frac{\text{R.E.} + 2545 \times \text{B.H.P.}}{3415 \times \text{K.W.}} \qquad \text{e.g.: C.O.P.} = \frac{20{,}900 + 2545 \times 3.65}{3415 \times 3.30} = 2.68 \text{ at 112F \& 0F.}$$

Speed	730	820	920
C.O.P. (112F & 0F)	2.68	2.62	2.60

Temperature of Suction Gas = 65F.
Liquid refrigerant taken at saturation.

[1]Refrigerating effect in thousands of Btu per hour.

Table 4 — Coefficient of Performance

Saturated Condensing Temp., F	Line Number	Saturated Suction Temperature, F							
		50	40	30	20	10	0	—10	—20
90	1	13.75	11.00	9.17	7.86	6.87	6.11	5.50	5.00
	2	12.24	9.71	7.96	6.76	5.83	5.13	4.50	4.06
	3	9.42	7.36	5.99	5.11	4.34	3.84	3.38	3.03
	4	6.51	5.43	4.64	4.01	3.53	3.15	2.64	2.36
	5	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
96	1	12.09	9.93	8.42	7.32	6.47	5.79	5.25	4.79
	2	10.75	8.61	7.19	6.19	5.39	4.78	4.27	3.84
	3	8.14	6.46	5.43	4.65	4.03	3.56	3.19	2.85
	4	6.03	5.05	4.37	3.80	3.38	2.99	2.53	2.29
	5	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
102	1	10.81	9.06	7.81	6.85	6.11	5.51	5.02	4.61
	2	9.54	7.80	6.61	5.74	5.01	4.47	4.00	3.65
	3	7.26	5.82	5.01	4.29	3.76	3.35	3.01	2.72
	4	5.58	4.76	4.13	3.60	3.21	2.86	2.45	2.21
	5	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
107	1	9.95	8.46	7.36	6.52	5.85	5.30	4.85	4.46
	2	8.60	7.15	6.18	5.37	4.75	4.24	3.81	3.47
	3	6.48	5.36	4.66	4.03	3.55	3.19	2.88	2.61
	4	5.17	4.46	3.92	3.43	3.07	2.78	2.37	2.14
	5	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
112	1	9.23	7.94	6.98	6.22	5.61	5.11	4.69	4.33
	2	7.89	6.64	5.74	5.03	4.48	4.03	3.63	3.32
	3	5.95	4.99	4.34	3.81	3.37	3.02	2.74	2.49
	4	4.83	4.20	3.71	3.28	2.96	2.68	2.30	2.08
	5	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
117	1	8.61	7.49	6.63	5.95	5.39	4.93	4.54	4.21
	2	7.21	6.20	5.40	4.76	4.26	3.85	3.49	3.19
	3	5.51	4.66	4.08	3.61	3.19	2.88	2.63	2.39
	4	4.53	3.98	3.54	3.16	2.86	2.34	2.34	2.03
	5	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00

Line 1—Coefficient of Performance of Carnot Cycle. (Highest conceivable, but actually unattainable)

Line 2—Coefficient of Performance of Freon-12 Refrigerant. These figures are for isentropic compression in a compressor without mechanical losses, operated by a motor 100% efficient; no super-heat; no sub-cooling.

Line 3—Coefficient of Performance of Freon-12 Refrigerant. These figures are for compression in a compressor having 85% over-all efficiency, operated by a motor 85% efficient; 10 degrees super-heat; 10 degrees sub-cooling.

Line 4—Coefficient of Performance of a commercial compressor.

Line 5—Coefficient of Performance of resistance-type electric heater.

refrigerating systems of the future, should be sought somewhere between the figures of lines 3 and 4 in Table 4. Below line 4, lies restraining conservatism and above line 3, lies wishful thinking.

In using the figures in these tables for the study of a heat-pump for heating purposes, it must be borne in mind that the condensing temperature will be from 10 to 15 degrees above the hot air discharge temperature, and the suction temperature will be from 20 to 25 degrees below the outside air, or from 15 to 20 degrees below well water temperature, depending upon whether an air-to-air unit or a water-to-air unit is under consideration.

It should be emphasized that all of the accompanying tables give coefficients of performance for the compressor unit alone. That is to say, wherever "energy input to the system" has been considered, only that energy which goes to the compressor motor, has been included. If the energy input to auxiliaries, such as evaporator-blower motor, condenser-blower motor, and controls, were to be considered in computing an overall coefficient of performance, the results given in the accompanying tables would have to be decreased, considerably.

Weather Chemistry

Why does New York City feel colder when the temperature is around freezing than does an inland city where the air is dry and the temperature even lower? Dr. J. A. Babor of the College of the City of New York answers the question. "In cold, humid air," he explains, "a higher proportion of water molecules diffuses through the clothing, and a greater number of replacements occur than in the case of oxygen and nitrogen. As a result, more heat is removed from the body by water molecules in a cold humid climate than by the less frequently replaced oxygen and nitrogen molecules of a dry climate." To be comfortable in cold, clammy New York, Dr. Babor advises, wear a raincoat or some other impervious garment and thus prevent the diffusion. You can even work up a perspiration that way.

Solar House Heating— A Problem of Heat Storage

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In recent years, several solar houses have been built and tested, using various means for solar heat collection. As the efficiency of solar heat collection diminishes at increasing temperatures, a moderate collection temperature is essential. It has been shown that a relatively short-term heat storage is sufficient. A new method is described, using the heat of fusion of chemical compounds for solar heat storage, collecting the heat at a moderate, constant temperature with high efficiency. An experimental structure has been built at M.I.T. for comparative tests with various solar heat storage methods. Studies were sponsored by Solar Energy Conservation Research Project.

SOLAR house experiments conducted in the past may be classified according to their methods of collecting solar energy.

Collector	Collection Temp., F	Solar Heat Storage
1. Roof type: black sheet metal with water pipes, covered by glass panels	Above 120	Hot water stored in large basement tank
2. Roof type: black surface, with glass panels	Above 110	Heated air circulated to rooms and stored temporarily
3. Typical solar house: large south-facing windows admitting solar energy to the rooms	70 — 80	Interior of the room stores heat temporarily

Roof-type Collector with Hot Water Storage

An experimental structure of this type was built at M.I.T. in 1939 in connection with the Godfrey L. Cabot Solar Energy Research Project. A blackened copper sheet collector of 360 sq ft area was mounted on the roof, behind three air-spaced glass plates in insulated boxes. Copper tubes, soldered to the copper sheet, circulated water to a 17,000-gal insulated tank occupying the entire basement. The experimental two-room structure was heated entirely by solar energy during two heating seasons. Equations were derived by Hottel & Woertz for the performance of the collectors with varying numbers of glass plates as a function of the outdoor and the collector temperature. These experiments were interrupted by the war and a complete analysis of the efficiency of the system and its economic aspects has not been published.

This solar heat collector operated throughout the year. A maximum storage temperature of 195F was reached during the summer, when the overall efficiency of the combined collector-storage system approached zero. The winter minimum temperature was about 125F. The possibility of storing summer heat for winter use was found definitely not economical.

Roof-type Glass Panel Collector, Circulating Air

G. O. G. Löf is conducting an experimental investigation of this type at the University of Colorado. About one-third of the test house is covered with the collector and approximately a 20% saving of fuel was obtained during one winter. With an average daily exit air temperature of 110F, an efficiency of 35 to 40% could be obtained, but at higher collector temperatures the efficiency decreased considerably. The conclusion was reached that the "use of the solar heat

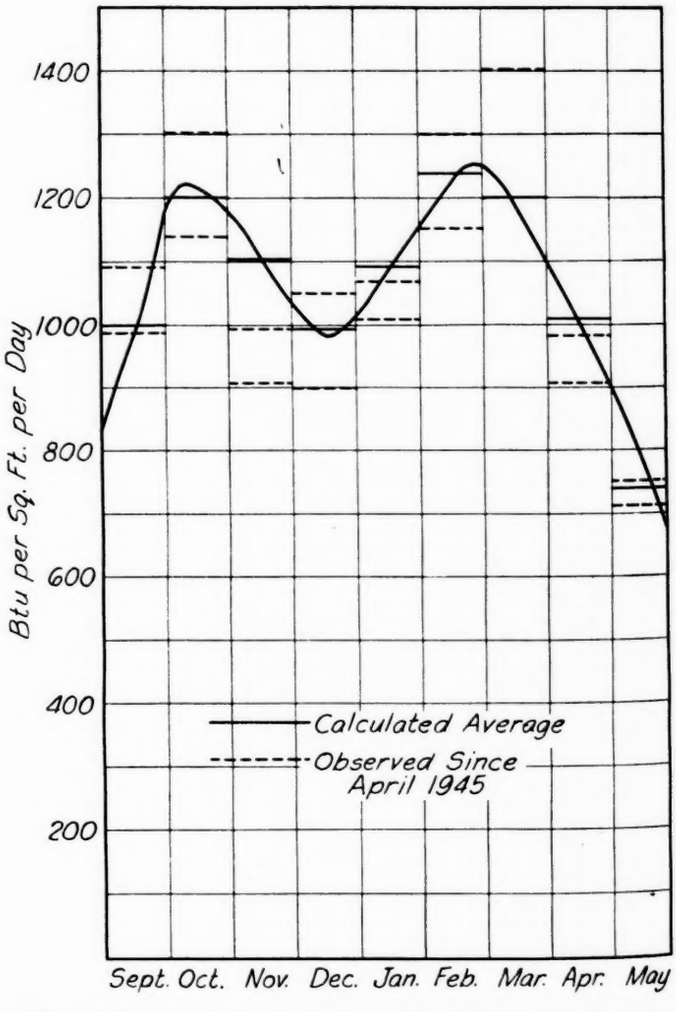

Fig. 1. Solar energy incident on a south facing vertical surface at Blue Hill.

collector without the auxiliary heat storage unit is probably impractical in most locations.

Solar Houses Using Large South-facing Windows

An experimental house of this type was built near Chicago, designed by G. F. Keck and sponsored by the Illinois Institute of Technology. The major part of the south wall of the house was occupied by air-spaced, sealed double-glass windows, transmitting solar energy into the rooms. The heating system of the house was of the radiant floor type. On sunny days the solar energy absorbed inside the rooms furnished supplementary heat and this was stored to some extent in the concrete floor. Due to the uncertainty regarding some heat transfer coefficients, the calculated heat saving was not ascertainable, but was possibly as high as 18%. During clear winter days the house became overheated.

Other solar houses were built according to the same principles and fuel savings up to 30% have been reported.

Large glass areas transmitting solar energy produce a "greenhouse effect." A historical summary of this effect has been published by K. Taylor and E. W. Gregg. The principles involved have been summarized in a University of Illinois bulletin.

Recent experiments at Purdue University by F. W. Hutchinson were designed to furnish comparable results. Two identical houses were built, one with conventional windows, the other with large south-facing windows. These tests are now in progress.

Summarizing conclusions of the preceding experiments, the following factors appear to be evident.

(a) It was possible to heat an experimental structure entirely with solar energy, using a relatively long-term heat storage, which is not economical.

(b) Fuel savings were obtained by using winter solar energy with practically no heat storage facilities.

(c) Relatively high temperature heat storage is quite inefficient and unnecessary.

(d) Relatively short term heat storage may be adequate in some southern locations.

(e) Large glass windows or glass-covered roof areas involve architectural problems. Large glass windows transmit excess daylight (glare) and restrict privacy (fishbowl effect) which may be undesirable.

Analytical Treatment of the Heating Problem

The following calculations were made for the vicinity of Boston, Mass., and can be applied to other regions where the amount of solar energy received is approximately the same and where the seasonal heat load (degree days) is comparable.

Amount of Solar Energy

The U. S. Weather Bureau publishes records of the total solar and sky radiation received on a horizontal surface at some 20 stations. Some of these data have been collected for over 20 years, and provide satisfactory average weekly values.

At Blue Hill (about ten miles from the center of Boston) the solar energy data, incident on a horizontal surface, have been recorded for the past 15 years. Irving F. Hand, in charge of Solar Radiation Research at the U. S. Weather Bureau, has been collecting additional data since March 1945, recording solar energy incident on a south-facing vertical surface at Blue Hills only.

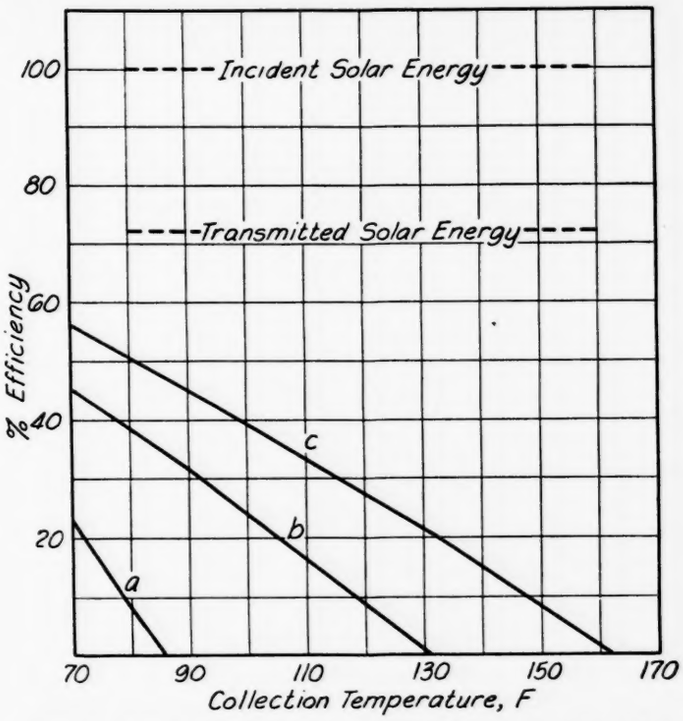

Fig. 2. Calculated efficiency of south facing vertical solar energy collectors at Blue Hill, using two glass plates.

Dec.-Jan. average outdoor temperature, 30F. Solar energy is 1000 Btu per sq ft per day. Outdoor losses—a. 24 hr; b. 8 hr, diminished loss 16 hr; c. 8 hr, no loss 16 hr.

Using these data, E. M. Brooks has established average values for vertical incidence. His data are shown as monthly averages in Fig. 1 and Table 1, together with the observed data*. Most of the observed data agree within less than 10% with the computed average, but differences up to 20% do occur.

Conclusion can be drawn that the amount of solar energy received during a month may generally be within 10% of the expected normal average, with occasional unseasonal deviations not exceeding 20% of the average.

Normal sunshine for the heating season (October to May) varies between 1,000 to 1,200 Btu per sq ft per day, received on a south-facing vertical surface.

Table 1—Solar Energy Incident on a South-facing Vertical Surface at Blue Hill

Averages Calculated by E. M. Brooks

Month	Btu per sq ft per day
September	1000
October	1210
November	1110
December	1000
January	1090
February	1240
March	1200
April	1010
May	740

*Data published through the courtesy of I. F. Hand.

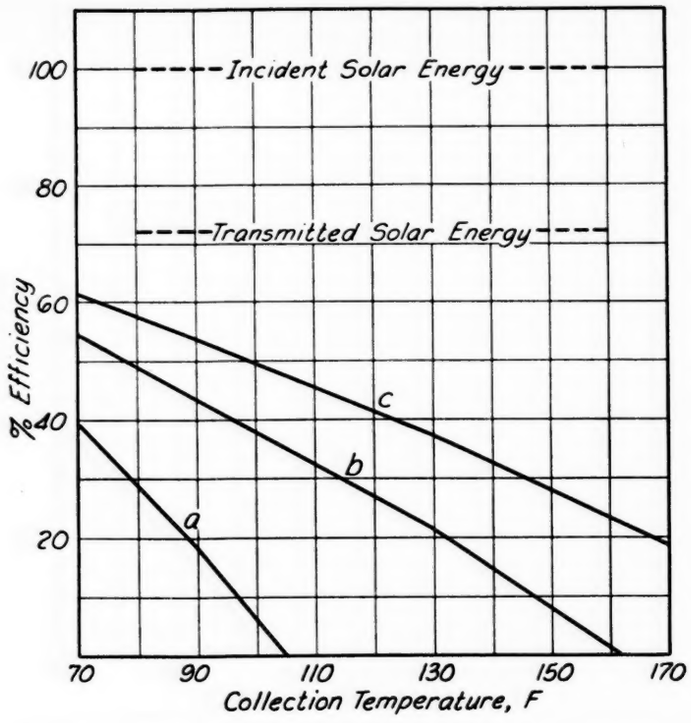

Fig. 3. Calculated efficiency of south facing vertical solar energy collectors, at Blue Hill, using two glass plates.

Dec.-Jan. average outdoor temperature, 30F. Solar energy on clear days is 1500 Btu per sq ft per day. Outdoor losses—a. 24 hr; b. 8 hr, diminished loss 16 hr; c. 8 hr, no loss 6 hr.

Only a part of the incident solar energy will be transmitted by two air-spaced glass plates. Hottel and Woertz's values for the transmissivity as a function of the angle of incidence have been used. The angle of incidence was obtained from the solar azimuth and altitude data. To carry out the computation it is necessary to know the average hourly values of solar energy. This has been done for the December-January average when 72% of the incident solar energy is transmitted by two air-spaced window-glass plates. During the other months the transmitted amounts of solar energy diminishes as the angle of incidence is progressively less favorable.

Efficiency of Solar Heat Collection

The efficiency of fuel-burning, house-heating systems ranges from about 50% for hand-fired, coal, heating systems to about 70% for gas or oil burning systems.

We now proceed to evaluate the amount of solar energy that can be collected on an absorbing surface placed behind two air-spaced vertical glass plates. We assume that 95% of the transmitted amount of solar energy is absorbed by the collector at a definite temperature. The collector will lose heat through the glass plates in proportion to the temperature difference between the collector and the outdoor air. We assume in these calculations that the collector does not lose heat from any other surface, except through the glass plates. The average outdoor temperature records of the Weather Bureau were used in combination with the overall heat loss coefficients for two glass plates for the outward-loss heat calculations. Obviously the difference between the transmitted amount and the outward-losses will represent the "net gain" in solar energy. The efficiency is defined as the ratio of the "net gain" to the total incident amount of solar energy*. The collector can operate in the following way.

Typical solar house—The collector is the interior of the house, normally at 70F. The glass transmits solar heat during the daytime (practically eight hours during the winter months) and loses heat during twenty-four hours. The overall heat loss coefficient U is about 0.52 for two vertical glass plates when the indoor temperature is 70F; U increases somewhat with the temperature.

Sun-wall collector with insulating partition — The collector is at 70F or higher. While the glass transmits heat during the daytime, it loses heat outwardly during this time. During the night an insulating partition is lowered between the glass and the collector, diminishing the heat loss. The insulating partition should be designed in such a way that when it is used the U value of the combined two vertical glass plates and partition is about 0.15. The average U will be 0.27 for 24 hours.

Wall or roof-type collector—This collector is insulated from the house with the collected solar heat circulated to a storage tank inside the house. The collector loses heat outwardly only during the daytime.

The efficiency of various collector arrangements has been calculated for the December-January average solar energy conditions at Blue Hill. The efficiency is summarized in Table 2 and Fig. 2.

Table 2 — Calculated Efficiency of Solar Energy Collectors at Blue Hill, December-January Average

Average incident solar energy, 1,000 Btu per sq ft per day
Solar energy transmitted (72% efficiency) through two glass plates, 720 Btu per sq ft per day

Collector Conditions	Collector Temp., F	Net Gain, Btu per sq ft per day	Efficiency, %
(a) Typical solar house: Energy collected 8 hours; heat loss 24 hours	70	230	23
	85	0	0
(b) Sun-wall collector with insulating partition: Energy collected 8 hours; heat loss diminished by insulating partition during 16 hours; U = 0.15	70	460	46
	90	320	32
	130	0	0
(c) Wall or roof-type collector: Energy collected 8 hours; heat loss during 8 hours	70	560	56
	90	450	45
	130	220	22
	160	0	0

These calculations show that the December-January solar heat collecting efficiency of the typical solar house is about 23%, provided the temperature of the rooms

*This net gain may be called the absolute net gain. A relative net gain may be defined as the difference between the transmitted solar energy and the outward loss from the solar energy collector, corrected for the outward loss from a conventional type of wall which is replaced by the solar energy collector.

is maintained at 70F. In an actual solar house, overheating will occur on clear winter days and heat will be wasted, diminishing the efficiency. With the insulating partition lowered at night, the collector operating at 90F shows an efficiency of 32%. The wall or roof-type collector operating only during the daytime may reach the same efficiency at a collector temperature of 110F.

On average clear days, the December-January amount of sunshine incident on a south-facing vertical wall at Blue Hill can be as high as 1,500 Btu per sq ft per day. This amount approaches the winter average for locations further south, and at higher altitudes. Due to the increased amount of solar energy, the efficiency will be higher, as shown in Table 3 and Fig. 3.

Table 3 — Calculated Efficiency of Solar Energy Collectors at Blue Hill for Clear Days, December-January Average

Incident solar energy, 1,500 Btu per sq ft per day
Solar energy transmitted (72% efficiency) through two glass plates, 1,080 Btu per sq ft per day

Collector Conditions	Collector Temp., F	Net Gain, Btu per sq ft per day	Efficiency, %
(a) Typical solar house	70	582	39
	90	268	18
	105	0	0
(b) Sun-wall collector with insulating partition	70	812	54
	90	664	44
	130	336	22
	160	0	0
(c) Wall or roof-type collector	70	912	61
	90	808	54
	130	576	38
	160	370	24

The increased efficiency on clear days is obvious when Fig. 2 and 3 are compared. It should be remembered, however, that on cloudy days the net gain and the efficiency will be zero unless more than a certain minimum or threshold amount of solar energy is received. These threshold values for average winter conditions are shown in Table 4.

Table 4 — Threshold Values for Average Winter Conditions

Collection Temperature, F	Daily Btu Per Sq Ft
70	168
90	270
130	500
160	710

A somewhat higher efficiency will be obtained during the months of November, March and April. During the months of September, October and May the efficiency will obviously reach its highest value as the outward heat losses diminish considerably.

Table 5 — Comparison of Various Low Cost Materials on an Equal Volume Basis

Material	Specific Heat Btu per Lb per F	Density Lb per Cu Ft	Heat Capacity of One Cu Ft, Btu per Cu Ft per F
Water	1.00	62.5	62.5
Iron, scrap	0.112	489	55
Concrete	0.27	140	38
Brick	0.20	140	28
Magnetite, iron ore	0.165	320	53
Basalt, rocks	0.20	180	36
Marble	0.21	180	38

Heat Storage Based on Specific Heat Effect

Water has been used most frequently as the medium of solar heat storage. Various other materials have been, or could be contemplated with the aim of decreasing the heat storage volume. Table 5 gives a comparison for various low cost materials on an equal volume basis. It shows that the amount of heat stored per cubic foot volume could not be increased by replacing water with any of the above materials.

Solid heat storage materials could be used advantageously because they eliminate the rather costly storage tank. A concrete wall may be used for both heat storage and as a structural part of the house, when adequately insulated. Such a wall could form a heating panel for the rooms.

The most serious difficulty encountered with the specific heat type of heat storage is the limited temperature range. Assuming that an efficiency of 30% is desired, the (b) type arrangement could be used with an average 30% efficiency in the temperature interval of 100—80F. One cubic foot of water would store in this case 1,300 Btu and one cubic foot of concrete only 1,000 Btu. With the roof type (c) arrangement an average efficiency of 30% could be realized in the temperature interval of 130—90F, or approximately twice the above values per cubic foot storage volume.

Heat Storage Based on Heat of Fusion

Heat of fusion, or heat of transformation, of certain chemical compounds has been suggested by the

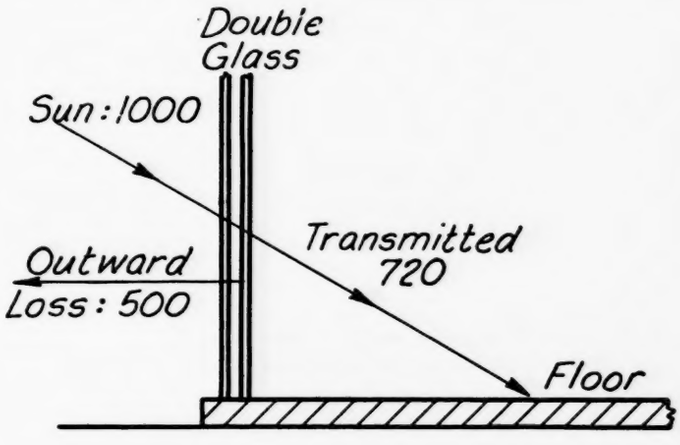

Fig. 4. Solar house with large windows.
Dec.-Jan. average conditions for the vicinity of Boston. Figures are in Btu per sq ft per day.

writer for solar heat storage purposes. Several low-cost chemical compounds are available which show a relatively large heat of fusion, melting in the vicinity of 90F. Suitable containers are used filled with the chemical compound and permanently sealed. They absorb solar energy directly, or solar heat may be transferred to them indirectly. Solar heat absorbed by the chemical compound below its melting point increases the temperature until the melting point is reached. Further heat is absorbed at the melting point without any appreciable increase in temperature, until all of the chemical compound is fused or transformed. Above the melting point the chemical compound absorbs further heat in accordance with its specific heat.

One of the chemical compounds suitable for this purpose is Na_2SO_4, $10H_2O$ (Glauber's salt). The heat of fusion or heat of transformation of this material is 104 Btu per lb; its density is 92 lb per cu ft. One cubic foot of Glauber's salt, therefore, could store 9,500 Btu at its melting point, which is about 90F (32.38C). At the melting point the Na_2SO_4. $10H_2O$ changes to Na_2SO_4 + $10H_2O$. Most of the Na_2SO_4 dissolves in the water of crystallization, leaving a part of the anhydrous salt as an insoluble residue. The stored heat may be transmitted to the rooms. As the heat is given off, the above process is reversed and solid Glauber's salt is formed again. The heat accumulation and depletion may be repeated indefinitely; the chemical is permanently sealed and never has to be replaced. Using arrangement (b), wall-type collector with insulating partition, the following comparison can be made between water and Glauber's salt as a heat storage medium, when both operate at an efficiency of 32% in the temperature interval of 100—80F;

Heat stored in one cubic foot of water	1,300 Btu
Heat stored in one cubic foot of Glauber's salt,	
Below its melting point	600 Btu
at the melting point	9,500 Btu
above the melting point	900 Btu
Total	11,000 Btu

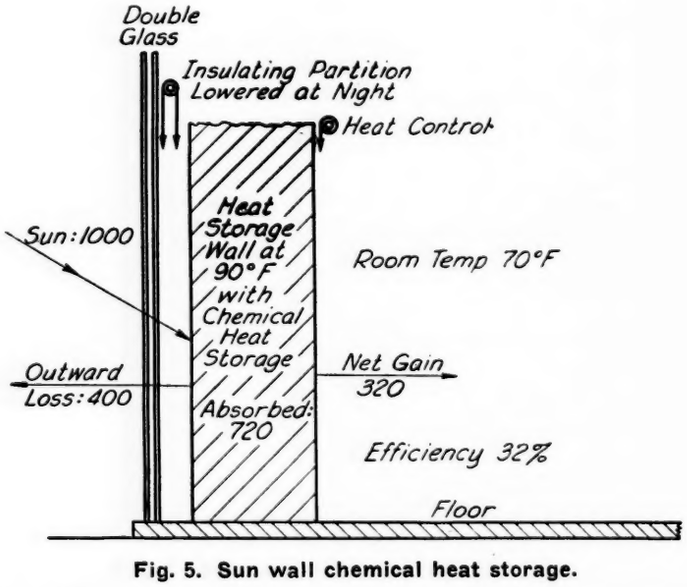

Fig. 5. Sun wall chemical heat storage.

Figures are in Btu per sq ft per day. Dec.-Jan. average conditions for the vicinity of Boston.

One cubic foot volume of Glauber's salt is therefore capable of storing about 8.5 times more heat than one cubic foot of water under identical conditions. Comparing the amount of solar heat stored in one cubic foot of water between 100 and 80F (1,300 Btu) with the amount stored by one cubic foot of Glauber's salt at 90F (9,500 Btu), the calculation shows that the Glauber salt can store 7.3 times more heat than water at its melting point. This calculation assumes that Glauber's salt melts and recrystallizes without any time or temperature lag. A correction factor may be introduced to account for such delays.

The heat-of-fusion type heat storage (Glauber's salt or other chemical compounds) compared with the specific-heat type storage (water, concrete, rocks, etc.) shows the following advantages:

The chemical compound can store at least seven times more heat than when using water, both being of equal volume and operating under identical conditions of collection efficiency.

The chemical compound collects solar heat at a constant, moderate temperature (90F), while naturally the temperature of water varies continuously.

Minimum or threshold value of solar energy that is collectable during average winter conditions is lower at 90F; therefore, the chemical compound can store additional solar energy on dull days when the higher temperature collector would not operate at all.

The cost of the chemical compound must be low. Impure Glauber's salt or "salt cake" may be used at a very low cost. The chemical compound must be confined within permanently sealed containers, the cost of enclosure being comparable to the cost of the water tank.

The possibility of internal container corrosion can be eliminated by using suitable corrosion inhibitors. Additional agents can be used to promote melting and recrystallization.

Features of Chemical Heat Storage

On the basis of calculations given, it is obvious that several designs could be applied to the chemical heat storage method. The various types are shown in Fig. 4, 5 and 6.

Sun-wall Collector with Insulating Partition. The south wall of the house is replaced by the *solar-heat storage wall,* Fig. 5, leaving openings for conventional windows. The exterior of the heat-storage wall is constructed of two glass plates. Containers filled with the chemical compound and permanently sealed are assembled like building blocks to form the heat-storage wall. An air space is left between the glass and the containers to accommodate the insulating partition, which is used at night to diminish the outward heat losses. The exterior surface of the containers is coated with a suitable coating to absorb solar energy (black paint). The interior surface of the containers serves as a heat transfer surface to distribute heat into the rooms; it is in effect a vertical radiant heating panel, operating at 90F, a suitable temperature for this purpose. Obviously the heat transfer into the rooms must be controlled to meet the variable heat load demand.

This can be accomplished with an interior heat-control device.

The component parts of the solar-heat storage wall are now described in greater detail.

Glass Plates. The glass plates are permanently sealed at the edges to form a single unit, with dry air between the plates to avoid condensation of water vapor and accumulation of dust. Obviously clear vision through the glass is not required; therefore, the glass is not plate or window glass but figured glass, which is available in relatively large sizes at a low cost per square foot. Since the pattern of the figured glass should not impede transmission of solar energy, the pattern should be rather shallow and might serve to improve the external appearance of the wall. Various patterns may be developed.

Insulating Partition. This partition is used to diminish the outward heat losses. The coefficient of heat transfer U is 0.52 for two glass plates with ½ in. airspace between 7/32 in. thick glasses. Calculations were given before for such an insulating partition operating during a night period of 16 hours, with $U = 0.15$ for the combination of two glass plates with insulating partition. Insertion of a solid insulating partition obviously produces two air spaces. Relatively thin insulating partitions made of insulating board, or made of blanket-type insulation with aluminum foil surface, could be used to attain a U value of 0.15 or even less. In the experimental structure built at M. I. T., window shades are used, coated with aluminum paint. The shades can be operated automatically with a solar-energy actuated device of the same type used for controlling street lights. In practice it was found that the insulating partitions should be raised in the morning and lowered at night, because the sky radiation is generally above the threshold value of collection during the daytime, even on cloudy days. During the summer, the insulating partitions are permanently closed, reflecting summer sunshine.

Containers. Requirements for containers were mentioned. Mass-produced containers filled with the chemical compound and permanently sealed could be designed as unit building blocks with interlocking features. The weight of containers should be limited to facilitate handling and shipping. Large containers, to be filled after they are installed, may be considered if economically satisfactory. The vertical dimension of the containers must be limited to prevent stratification of the chemical compound. Further development work must be carried out to determine the most efficient shape factor of the containers.

Interior Heat-control Device. Admission of heat from the solar-heat storage wall into the rooms must be controlled thermostatically to maintain a constant room temperature of 70F. The interior heat-control device may be in the form of curtains, shades or louvers, admitting heat into the rooms when opened. A fixed insulating partition may be used in the form of a ½-in. thick insulating wall board leaving an airspace behind the containers for the forced circulation of the room air. Floor and ceiling louvers can direct passage of the air.

Wall or Roof-type Collector with Chemical Heat Storage. Fig. 2 and 3 show that a higher calculated efficiency could be obtained with a wall or roof-type collector losing heat outwardly only during the daytime. To make such a collector feasible, its rear side must be thoroughly insulated, but it is obvious that the calculated efficiencies could not be fully realized. This collector could use air or any suitable heat transfer medium, circulating heat partly to the rooms and partly to a heat exchanger, transferring the excess heat to a storage tank or storage wall filled with the chemical compound. For this purpose a somewhat higher collection temperature is unavoidable, decreasing the collection efficiency.

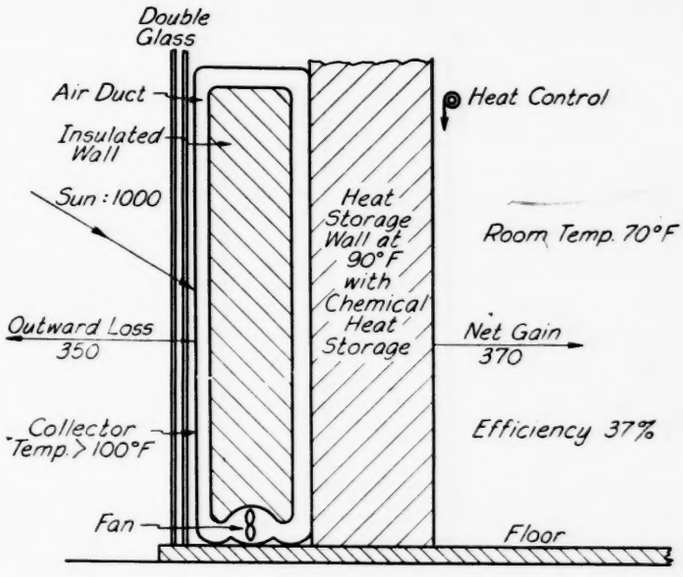

Fig. 6. Wall or roof type chemical heat storage.
Figures are in Btu per sq ft per day. Dec.-Jan. average conditions for vicinity of Boston.

Such a collector could be placed in the south wall of the house, behind two glass plates. The collector in this case would operate at less than 100F and would be in the form of a black-coated duct through which air is circulated to the heat exchanger surrounded with the chemical compound. An insulating wall is placed behind the duct. At night the combined glass, duct and insulation (Fig. 6) may show a U value as low as 0.06, and the December-January net gain in this case would be 370 Btu with an efficiency of 37%.

Heating Analysis for House in Boston Area

Let us examine the possibility of heating a house in the vicinity of Boston entirely with solar energy, using wall-type chemical heat storage.

Applying data presented, to the calculations for an actual house, we can answer the decisive questions:

What is the wall collector area?

How much heat storage is required?

As a yardstick we may use a hypothetical house in the vicinity of Boston, requiring 100 million Btu (1000 Therms) during the heating season. This amount could be obtained by burning about 1,000 gallons of fuel oil in a conventional furnace, at an efficiency of 70%.

The monthly distribution of the heat load is obtained from the degree-day data (1946 revised values). The monthly amount of solar energy is obtained from

Table 1. The heat load and the amount of solar energy are compared in Table 6 and Fig. 7.

Table 6 – Solar Energy and Heat Load in the Vicinity of Boston

Months	Degree Days	Heat Required Million Btu per Month	Average Solar Energy Btu per Sq Ft per Month		Collector Sq Ft
			Incident	Transmitted	
September	98	1.7	30,000		
October	338	5.8	37,500	21,000*	
November	651	11.1	33,300	23,000*	480
December	1000	17.1	31,000	22,300	770
January	1101	18.8	33,800	24,300	775
February	1022	17.5	34,700	23,000*	760
March	852	14.6	37,200	21,000*	700
April	538	9.2	30,300		
May	248	4.2	23,000		
Total	5848	100.0			

* Estimated.

According to Fig. 7, the amount of solar energy transmitted through 800 sq ft south-facing vertical surface would practically balance the heat load during the critical months of December, January and February. A large surplus of solar heat is available during September-October, with some probable surplus during the balance of the heating season.

The heat load of the house is distributed in the following way. The south heat-storage wall, at a temperature of 90F, has an average overall coefficient of heat loss U = 0.27 (U = 0.52 for 8 hours and U = 0.15 for 16 hours). The other surfaces of the house should be insulated in such a way that the overall heat loss coefficient is U = 0.12, including air changes.

From this data it is possible to calculate the dimensions of the yardstick house (using 100 million Btu for the heating season) assuming that the house is of rectangular shape with a flat roof. As the south-wall area should be 800 sq ft, a two-story house may be built, with the south-wall dimensions being 18 ft high and 44.5 ft long. A simple calculation shows that the depth of the house would be about 16 ft, representing a small-average four or five room house.

We may derive the same conclusions by using Table 2. The efficiency of a sun-wall collector at 90F is given as 32% of the incident solar energy for December-January. The net gain, therefore, will be 10,800 Btu per sq ft of wall collector. The resulting dimensions are the same as calculated.

It is obvious that the efficiency of the wall-type collector could be increased by improving the heat-insulating partition, inserted between the glass and the heat storage wall during the night to prevent the escape of heat.

It should be emphasized that conventional windows may be used in the south wall. Such windows admit heat directly into the rooms, averaging an efficiency of 45% as daytime heat collectors. At night the windows would be covered with heat insulating partitions.

Amount of Heat Storage Needed

Calculations are based on the assumption that the monthly average solar energy is equal to or greater than the monthly heat load. An overnight heat storage would, therefore, be sufficient if the exact average daily amount of solar energy could be received each day to balance the exact average daily heat load. Such weather conditions will occur very rarely. Clear days, with above average solar energy, alternate with partly cloudy and cloudy and dark days. A survey was made of the Weather Bureau's records for Blue Hill, tabulating the daily amounts of solar energy for December (14 years' records were available). It was found that during the past Decembers an average of 17 days were above-average sunshine days and only 14 days were below-average sunshine days. The records were further analyzed for sequences of below-average sunshine days. It was found that the longest period of successive below-average sunshine days lasted 6 days, occurring only once in 14 years. Similar 5-day sequences may be expected every second year. The conclusion may be reached that the storage of 6 average days of solar energy would prove to be sufficient to provide adequate storage for consecutive below-average sunshine days.

It was mentioned above that the actual monthly solar energy may be 10% below the average monthly amount. In a 30-day month this would require an

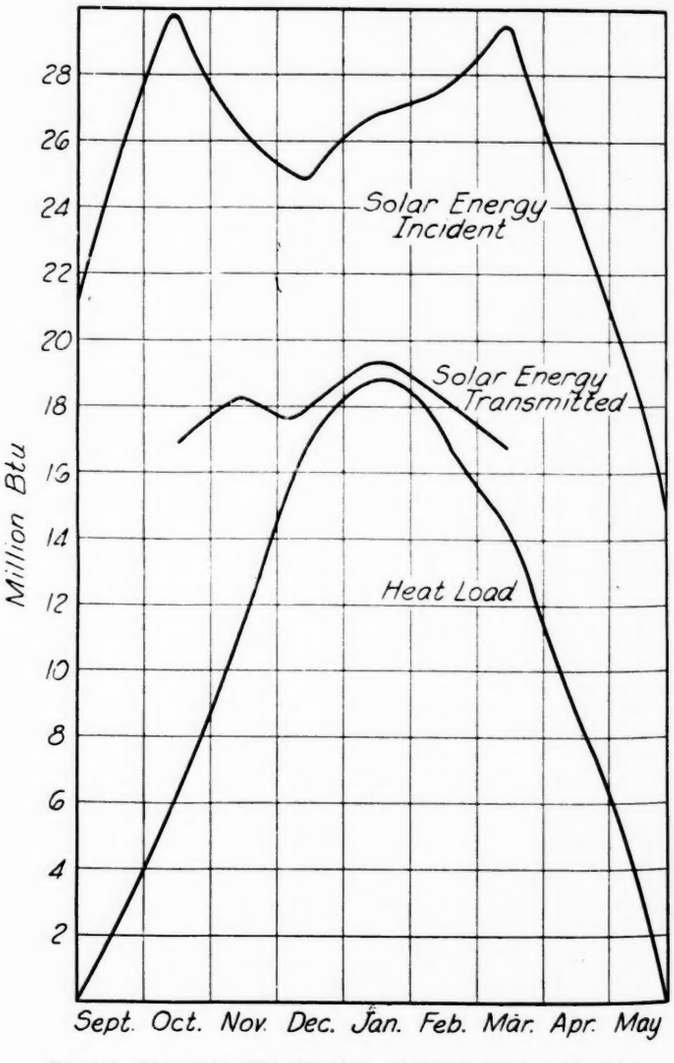

Fig. 7. Monthly distribution of heat load and solar energy at Blue Hill.

Heat load—seasonal 100 million Btu. Solar energy—incident on and transmitted through two glass plates of 800 sq ft area—vertical, facing south.

additional three average days of heat storage.

The writer believes that a total heat-storage capacity for 9 average days of solar energy would be sufficient for the vicinity of Boston.

During December-January the daily average solar energy transmitted through two glass plates is about 720 Btu per sq ft. Using Glauber's salt as the heat storage medium, an average day of solar energy (720 Btu per sq ft) may be stored in 7 pounds of Glauber's salt, or the 9 days' storage can be accomplished with 63 lb of Glauber's salt occupying a volume of 0.69 cu ft, in a container 12 by 12 by 8¼ in.

The conclusion can be reached that a 9-in. thick solar-heat storage wall, using Glauber's salt as the heat-storage medium, would provide adequate heat storage for the above conditions. Returning to the yardstick house, the total storage volume would be 600 cu ft and the total weight of the Glauber's salt would be 28 tons, consisting of 16 tons of water and 12 tons of anhydrous sodium sulfate or "salt cake" (cost approximately $240 unpackaged).

Calculated efficiency of a wall or roof-type collector as shown in Fig. 6 is about 37%. The greatest advantage of such a collector would be the absence of the insulating partition and there would be no need to operate and control it. Its disadvantage would be that air-circulating fans would operate continuously during the daytime and the rate of air-flow would have to be regulated rather exactly to limit the temperature of the collector. The increased efficiency of 37% would correspond to a January net gain of 12,500 Btu per sq ft. Using such a system of 800 sq ft area in the above-mentioned yardstick house, the dimensions of the house can be larger. A two-story house 18 ft by 44.5 ft long and 19 ft deep could be built with this method as compared with a depth of 16 ft for the method according to Fig. 5.

The calculations show the feasibility of an entirely solar-heated house in the vicinity of Boston. Locations receiving more solar energy and requiring lower heat loads, would naturally require a smaller heat collector and a smaller storage volume.

It should be emphasized again that these calculations refer only to the hypothetical yardstick house in which the monthly distribution of solar energy and heat load is in accordance with Fig. 7 and Table 6. The following assumptions were made:

(a) The outward heat losses from the glass do not exceed the allowed limits.

(b) The interior heat control operates efficiently and prevents overheating of the rooms and the wasting of stored heat.

(c) The containers are designed with the most favorable shape factor, preventing possible local overheating, stratification of the chemical compound, and inadequate heat transfer.

These variables influence the performance of the solar heat storage wall. Their most favorable design, effective installation and control is the subject of further investigations.

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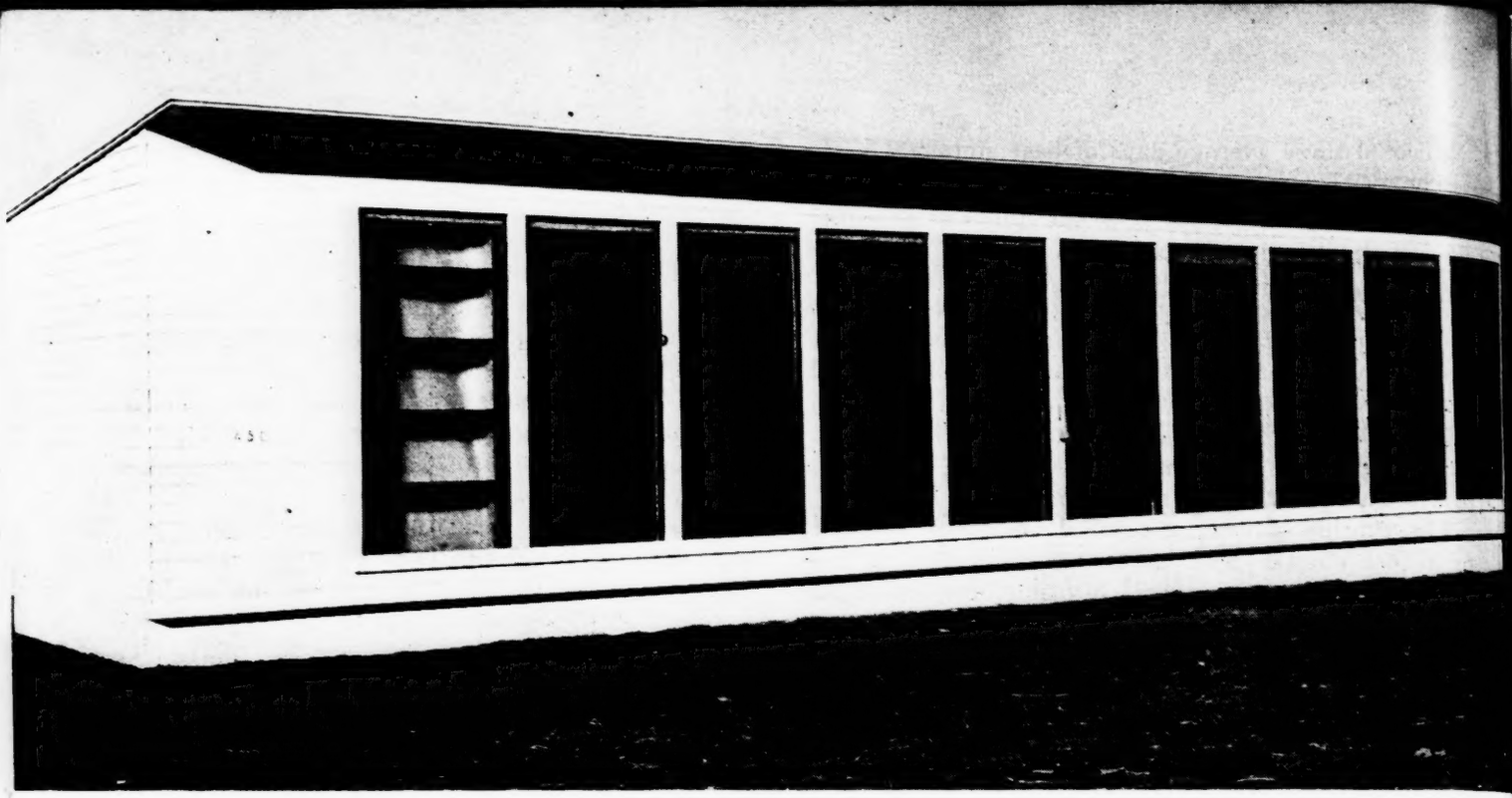
Aluminum Paint for Heating and Ventilating Systems

Aluminum paint, now produced in three specific grades, has been used extensively in heating and ventilating work to preserve and brighten galvanized metals. Serving both as a primer and a means of improving appearance, aluminum paint is pretested against cracking or peeling from heat or corrosion. In addition, it protects the galvanized coating on ducts from heat and moisture.

Aluminum paint, according to Alcoa, provides a material which can be used on practically any paintable surface. Tests show it has durability that equals or betters that of most darker colored paints; that it is superior to other paints of equally light color.

Aluminum paint has been materially bettered over a 15 to 20-year period. Its light-reflecting properties make it an excellent interior paint because it achieves practically maximum reflectivity in one coat. Aluminum paint reflects heat as well as light, its heat reflecting power being generally given as 70%. Because of its high moisture-proof power and its ability to withstand washing as well as constant exposure to steam, fumes, and other corrosive agencies, it is now being widely used on galvanized metal surfaces.—*H. F. Johnson, Aluminum Company of America.*



Exterior, solar energy research structure at M.I.T.

Solar Heat Test Structure at M. I. T.

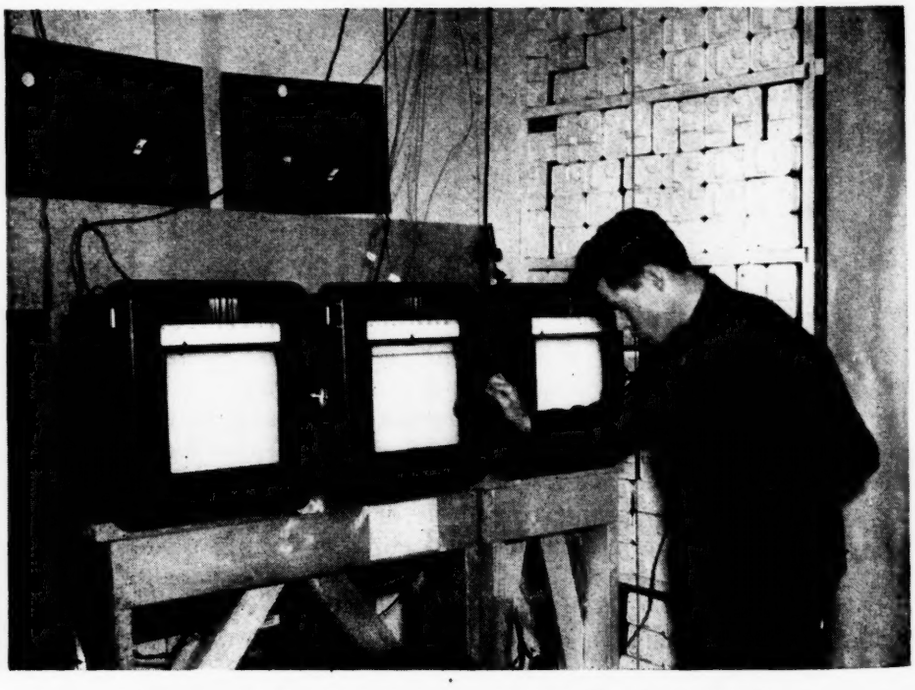
F. N. HOLLINGSWORTH

MASSACHUSETTS Institute of Technology engineers are conducting a novel study to determine whether a house or small commercial building can be adequately heated from solar radiation alone, in any kind of a climate, and whether this solar heat can be stored for future use, and for what length of time.

The Institute describes this project as "part of a research program for heating houses with only solar energy, to save critical material and fuel by eliminating conventional heating systems. "Data obtained from the experiment during cold months will be of value in designing homes and other buildings with a view to utilizing solar heat in a practical way.

The test building is a one-story structure with roof and floor insulated by 4 in. of rock wool. Exterior walls on three sides are also insulated with 4 in. of rock wool. The roof has considerable overhang to shield off rays of high summer sunlight. There is no foundation or cellar, the structure resting on wood piers two feet above the ground. Interior space is divided into seven cubicles with celotex walls, each cubicle being a separate study or experiment. Walls between are also insulated by 4 in. of rock wool, passage from one cubicle to the others being through well insulated doors. Entrance at rear is by similar doors.

Object of dividing the 14 x 50-ft interior is to concentrate seven years of research into one year. Each cubicle is 4 ft wide and 8 ft high, and is considered a separate project with a different heat storage arrangement, although all are connected to master recorders. All but one have double glass on the south front, that one having triple glass, all of it 7/32 in. thick. The south wall was chosen because it more nearly provided a flat surface collector at the sun's lowest angle in



Checking solar energy release on recording instruments.

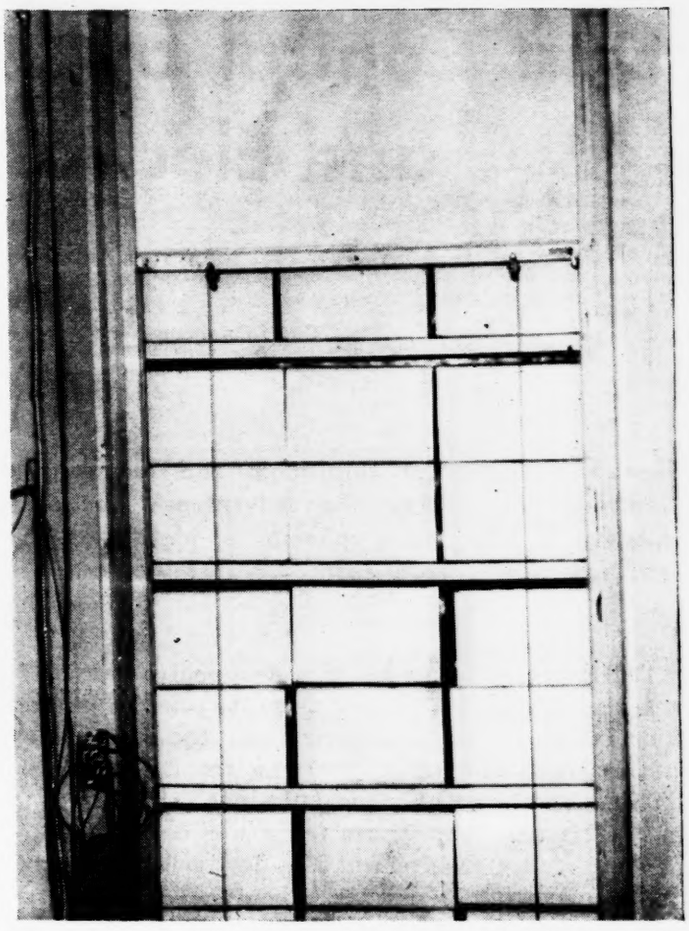
winter. The sun's rays can penetrate through the glass but they cannot readily escape. Since the glass does not prevent completely the outward flow of heat through it, the incoming heat would gradually seep outward were it not for a solid wall of heat absorbing containers of various sizes. These containers set close like bricks behind the glass and they transmit the captured heat to a chemical sealed within the containers. When the room temperature cools on sunless days or at night, the captured heat begins to flow back into the rooms, the containers acting together as a solid wall radiator.

Each cubicle has a set of double shades, operated automatically by small motors so the shades can be lowered at night, or on sunless days to conserve the stored heat. All sections have louvers or ducts with small fans to drive the stored heat out into the cubicle. Thermostats control fan operation. Containers can be of different sizes, according to a predetermined formula, and the chemical in them can be varied. It has been found that some of the walls have stored heat within themselves for several days. Some of the cubicles were built to store heat for only one day. On dull, chilly days an electric sunbowl is provided to keep temperature at 70F in the sections without sufficient solar heat storage. This auxiliary heat is measured. The outside wall is ordinary window type glass, of the kind known as "figured" or obscure. This type of glass is available at the lowest cost in large sizes which is the reason for using it. It is as satisfactory as window glass for transmitting the solar rays to the containers behind the glass.

A chemical used in the containers melts at 90F, a large amount of heat being absorbed without raising the temperature. At 90F, 100 Btu are absorbed per pound of chemical. The containers are painted black on the outer surface toward the glass, and white on the room side. The new method of using containers filled with chemicals to absorb and to store the solar heat is the result of research by Dr. Maria Telkes of the Institute and her patent application has been assigned to the Institute. Dr. Telkes is consultant on the project. Continuous automatic temperature records register the temperatures of the cubicles. The amount of solar radiation is measured by means of a standard pyrheliometer and a micromax recorder kept in a larger cubicle, which is also used as an office. Continuous readings are provided, even a passing cloud shows on the recording sheet. Being automatic, the readings and recordings are continuous, even when no one is on duty at the house.

While it is perfectly feasible to arrange for solar heating of houses or other structures, including small business and commercial buildings, the paramount question is whether such a system can be installed and be economical as compared with conventional heating systems. Therefore launching of this project does not constitute Technology's endorsement of the idea.

In attempting this novel system, Technology is trying to find the least costly combination of heat collector and storage unit. It is not yet known whether the present system should be designed for fairly long-time storage of solar heat and complete freedom from conventional heating, or whether the most economical system should provide for more than overnight heat storage, with an auxiliary heating system for sunless days.



Insulating curtain partially drawn over one wall section showing cans of chemical used in storage of sun's heat.

The committee having overall charge of the project comprises Professors L. B. Anderson of the Department of Architecture, chairman; Albert G. H. Dietz of the Building Engineering Department; and H. C. Hottel of the Department of Chemical Engineering. Paul Witherall of the Building Engineering Department is in direct charge of the research in the laboratory.

The present project, as was the one conducted by M.I.T. before the war, is part of a general program at Tech of research into means of utilizing solar energy, undertaken as a consequence of a gift to the Institute from Godfrey L. Cabot in 1937. Income from that gift is being used to study methods of improving use of solar energy.

"We think it is possible," says Professor Anderson, "to design a house which would receive on the south wall alone more than half enough solar heat to warm the whole house in January. And there are probably many places in the country where the south wall would take care of any winter condition.

"Modern architecture has already adjusted itself to the sun, and people have found that solar heat received through glass walls has compensated for a great deal of lost artificial heat. Double panes of glass reduce the heat loss and triple panes would be even better, though they send up the cost of construction."

Dust Control by Combined Centrifugal and Electrostatic Principles

E. H. R. PEGG
Chief Engineer, The Aerotec Corporation, White Plains, N. Y.

The combination of centrifugal and electrostatic precipitation utilizes the advantages of each method in effecting a collector of high efficiency and low maintenance cost.

THE first requirement of a dust control system is the removal of the dust from its point of generation before it can be dispersed into the surrounding atmosphere. The rules governing the design of the hoods, the volume of air and the size of the ducts to meet this requirement, are fairly well established.

A secondary requirement is a dust collector for removal of the dust from the exhaust air. The degree of dust collection required may be governed by such things as the character of the dust, state and local ordinances and whether the air is to be discharged outdoors or returned to the room.

Aside from the collection efficiency, a very important factor to be considered in the selection of a dust collector is the effect it has upon the operation of the system. Any collector whose resistance increases as dust accumulates will cause a gradual decrease of air volume with a consequent loss of efficiency of pick-up at the hood. The greater the dust loading or the higher the efficiency of the collector, the more rapid is this decrease. The result is a system which is either ineffective or requires excessive maintenance to make it effective. Therefore, a prime requirement of a dust collector should be that it maintains a practically constant resistance during the accumulation of a considerable quantity of dust.

Many types of dust collectors are in use today. Generally speaking, they can be divided into the following classifications:

(1) Viscous type filters where dust is trapped and retained on surfaces coated with adhesive.
(2) Dry type filters such as bags or mats of cloth, felt or treated paper which depend upon a screening action to remove dust from air.
(3) Centrifugal separators in which dust is thrown out of the air by centrifugal force imparted by whirling the air at high velocity.
(4) Combinations of two or more of the above types.
(5) Electrostatic precipitators in which the dust particles are electrostatically charged and precipitated from the air.

Each type has its advantages and disadvantages. The first two have the advantage of low first cost and the disadvantage of increasing resistance with dust accumulation. Centrifugal collectors have the advantages of constant resistance and high dust storage capacity and the disadvantage of low efficiency on very fine particles. The electrostatic precipitator has the advantage of very high efficiency, particularly on very small particles, and the disadvantage of low dust holding capacity.

This immediately suggests the combination of the centrifugal collector and the electrostatic precipitator to combine the advantages of both and to overcome the disadvantages of each. The result is a dust collector of very high efficiency, high dust storage capacity and a constant resistance which assures adequate air flow at the hood at all times.

The electrostatic method of removing solid or liquid particles from gases has been in use for many years. Until recent years, however, it was used principally for the cleaning of stack gases and could not be used for the cleaning of ventilating air because of the generation of large amounts of ozone and oxides of nitrogen which made the air unfit for breathing. About ten years ago, the principles were applied in such a way as to develop a precipitator of very high efficiency, even on smoke particles as small as .1 micron, and which generates negligible amounts of ozone and oxides of nitrogen so that it can be used for the cleaning of ventilating air.

The principle of operation of this new precipitator is shown on Fig. 1. It is, essentially, a two-stage precipitator in which the dust particles are first charged in an "ionizing" electrostatic field, and then precipitated on charged metal plates which constitute a uniform or "non-ionizing" field. When these plates be-

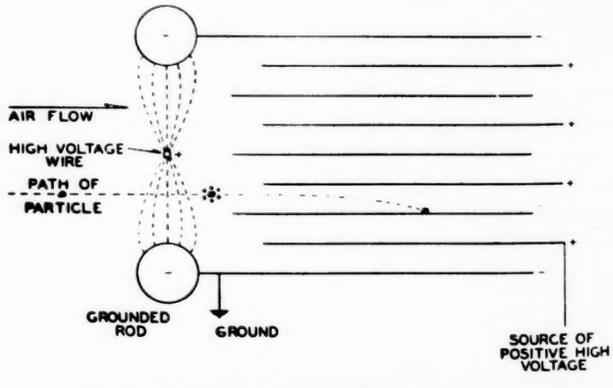


Fig. 1. Principle of operation of two-stage electrostatic precipitator.

Table 1—Performance of Centrifugal Dust Collector with Testing Sample

1 Particle Size, Microns	2 Percent of Sample by Weight	3 Average Size, Microns	4 Collector Efficiency, Percent	5 Percent Collected by Weight	6 Relative Weight	7 Number of Particles per Cubic Foot	8 Number of Particles Collected per Cubic Foot
+ 420	7.7	420.	100.0	7.7	72,000,000.	9	9
+ 175 — 420	33.6	298.	100.0	33.6	26,300,000.	107	107
+ 150 — 175	9.9	163.	100.0	9.9	4,300,000.	192	192
+ 75 — 150	22.1	113.	100.0	22.1	1,440,000.	1,280	1,280
+ 40 — 75	19.0	58.	100.0	19.0	194,000.	8,200	8,200
+ 30 — 40	3.7	35.	100.0	3.7	42,900.	7,200	7,200
+ 20 — 30	1.7	25.	100.0	1.7	15,600.	9,100	9,100
+ 15 — 20	0.8	17.5	98.0	0.79	5,360.	12,400	12,200
+ 10 — 15	0.4	12.5	95.0	0.38	1,950.	17,100	16,200
+ 7.5 — 10	0.35	8.75	90.5	0.31	670.	43,500	39,400
+ 5 — 7.5	0.25	6.25	85.0	0.21	244.	84,000	71,300
+ 4 — 5	0.17	4.50	78.0	0.13	91.	156,000	122,000
+ 1 — 4	0.18	2.50	63.5	0.12	15.6	960,000	610,000
+ 0 — 1	0.15	0.50	30.0	0.05	.125	100,000,000	30,000,000
Totals	100.00			99.69		101,249,088	30,897,188

come loaded with dust, it is necessary to clean them in some manner such as washing, scraping or vibrating them.

The reduction of ozone generation and the high efficiency are obtained at a sacrifice of dust holding capacity. Consequently, this precipitator has been used extensively for air cleaning in ventilating and air conditioning systems where the dust concentration is extremely low, but its use with high dust concentrations has been limited by the fact that the frequency of plate cleaning made it impractical for this application. The obvious solution is to precede the electrical precipitator by a mechanical collector to remove the larger particles which constitute the greater part of the bulk of the dust, leaving only the fine particles to be collected by the precipitator.

Since the object of the pre-cleaner is to reduce the frequency of cleaning of the precipitator, it is important that the efficiency of the pre-cleaner be high. This point can best be illustrated by considering the in-

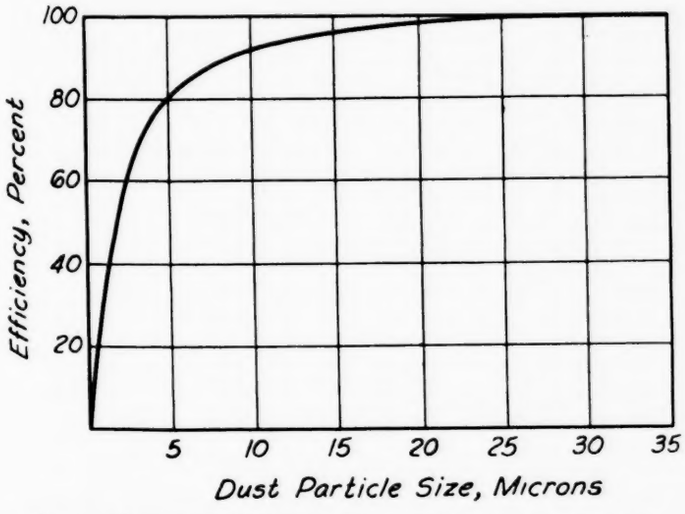


Fig. 2. Curve of efficiency vs. particle size of tubular centrifugal separator.

efficiency of the pre-cleaner. A pre-cleaner which is 98% efficient will pass only half as much dust to the precipitator as one which is 96% efficient, and consequently the cleaning of the precipitator will be only half as frequent in the former case as in the latter.

Recently, there has been developed a centrifugal collector of the tubular type which employs tubes of much smaller diameter than conventional tubular collectors with a resultant increase in efficiency and decrease in space required.

The efficiency of this collector on dust particles of various sizes is shown in Fig. 2. It will be noted that the efficiency on particles greater than 22 microns is 100%, and that the curve does not drop sharply until the particle size decreases to about 5 microns. The exceptional efficiency of this collector enables it to be used alone in many dust control systems. The construction of a unit dust collector for this purpose is shown in Fig. 3.

It it common practice in some localities to rate the performance of a dust collector on the dust count at the outlet in millions of particles per cubic foot of air rather than on the basis of weight efficiency. This is particularly true if the air is to be returned to the room. Industrial dusts are usually constituted of particles covering a wide range of sizes. There is such a vast difference in the respective weights of a coarse particle and a very fine particle that often the greater percentage of the number of particles represent only a small fraction of the total weight of the dust. Consequently, a dust collector may be very highly efficient on a weight basis and still be unacceptable on a particle count basis.

To illustrate these and other points, Table 1 was prepared. The particle size analysis in columns 1 and 2 is that of a dust used for testing unit dust collectors. The figures in column 4 were taken from the curve of Fig. 2. Column 5 is the product of columns 2 and 4 and the sum represents the weight efficiency of the

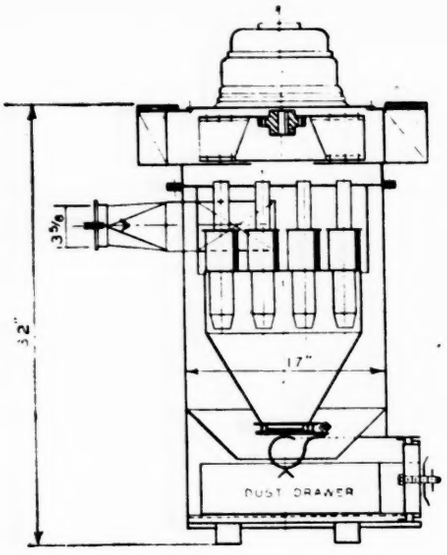


Fig. 3. Centrifugal unit dust collector, 300 cfm capacity.

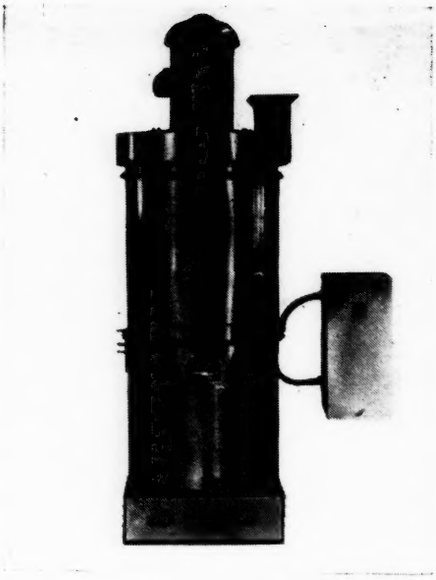


Fig. 4. A 600 cfm combined centrifugal-electrostatic unit dust collector.

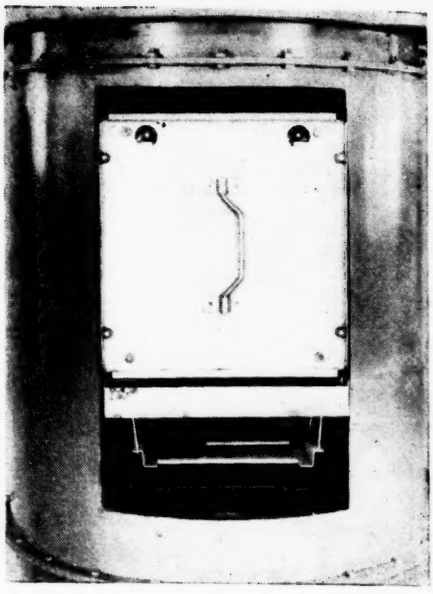


Fig. 5. Unit of Fig. 4 with cover removed to show electrostatic section.

collector. Column 6 shows the weight of one particle of each of the sizes shown in column 2 relative to the weight of a 1-micron particle of the same density. To obtain column 7, it was assumed that the dust concentration entering the collector was 2 grains per cubic foot and that all particles were of a uniform specific gravity of 3. Column 8 is the product of columns 2 and 7 and the sum of column 8 divided by the sum of column 7 is the efficiency of the collector on a particle count basis.

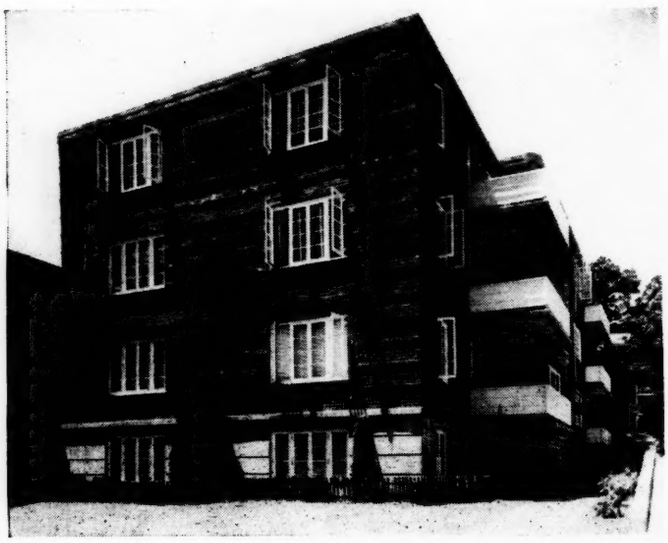
It is immediately apparent from this table that in spite of the almost perfect collection by weight, the dust count at the outlet is still some 70 million particles per cubic foot. This great discrepancy between the weight efficiency and the particle count efficiency is easily understood by examination of column 6 where a 420-micron particle is shown to weigh 576 million times as much as a one-half-micron particle.

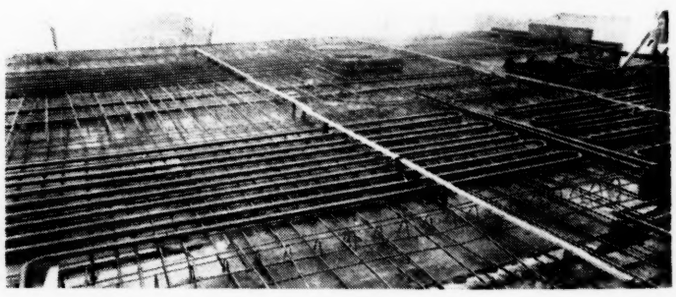
It is also apparent that for an industrial dust which did not contain any considerable quantity of particles below 5 microns in size, the collector shown in Fig. 3 would suffice, but that when these extremely fine particles are present, an after-cleaner of high efficiency on fine particles is required if a low outlet dust count is desired.

For the latter application, the electrostatic precipitator can be inserted between the centrifugal collector and the fan of Fig. 3, to make a combined centrifugal-electrical unit dust collector shown in Fig. 4. The electrical collector is removed from the casing for cleaning as shown in Fig. 5.

This combination presents a collector which has a high efficiency on particles of all sizes and has a practically constant resistance which insures adequate air flow in the system at all times. Because the bulk of the dust is deposited in a drawer where it is easily disposed of, the maintenance of the unit is a minor item.

Radiant Heated Apartment





One of the first apartment buildings to use floor-type radiant heating throughout was completed last summer at Cincinnati, Ohio, according to an A. M. Byers Co. report. The brick and concrete structure is three stories high and includes 14 apartments. Approximately 8 tons of iron pipe were used for the heating coils and mains imbedded in concrete floors.

What Goes On in a Typical One-Pipe Steam Heating System

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JUST what goes on in a heating system is one of those things which takes time to figure out and is therefore most interesting when analyzed for a typical case. Where does the air hang out in a one-pipe steam system; how much effect does its volume have on the heating system; and what is the effect of radiation of varying types, weights and air volumes in the same installation? What is the effect of the weight of various metals and the water within the system? Such factors are all considered within this article, together with the effect of more or less radiators turned on, raising of boiler water line, size of oil burner nozzle, size and length of pick-up and normal loads, piping tax, lower space temperatures, aquastat setting, time required for cooling off, effect of vacuum air valves and of stored heat.

The system described and illustrated has been literally taken apart main by main, risers from mains, radiators of copper from those of cast iron, while the air capacities and weights are separately listed. It is a one-pipe steam heating system in the writers home and is presumably typical for a fairly good system. The given figures are for conditions assumed as representative and are for the conditions stated, though such conditions do not prevail at all times. It is believed, however, that the reasoning is logical and that the points brought out at least show the consequence or importance of certain factors and do not leave their effect upon the average heating system open to as much conjecture as is usually assumed. The article offers nothing new to oil burner engineers, but it does present a concrete case to engineers who have no time for figuring these things out in such detail.

Referring to Table 1, each of the copper radiators varies but slightly from an average of 0.04 cu ft of air content and each branch connection to first floor radiators contains about the same amount of air, whereas the second floor copper radiators with connecting piping each averages a total of 0.15 cu ft or 100% more of air to be vented by their air valves due to the air in the five risers to which they connect. The cubic feet of air contained within the radiators per 100 sq ft of heating surface is 0.13 for copper, 1.18 for modern slenderized cast iron and 1.76 for the older cast iron tubular type.

The air in the mains is 4.7 cu ft while the steam space has a volume of 4.5 cu ft where air may or may not be present when the fire starts, depending upon how long since the burner last operated. There are three quick vent valves on the mains which vent the

Table 1—Basis for Computing Load

	Heating Surface, Sq Ft	Total Interior Volume of Air, Cu Ft	Total Weight, Pounds		Per 100 Sq Ft of Heating Surface	
			Metal	Water	Volume, Cu Ft	Weight, Pounds
11 Copper Radiators	331.2	0.43	320		0.13	97
5 Arco Radiators	146.9	1.74	605		1.18	412
2 Corto Radiators	65.0	1.14	358		1.76	550
Bare Main "A"	42.0	1.50	128			
Bare Main "B"	30.0	1.00	188			
Bare Main "C"	23.0	2.20	358			
Covered Main "C"	EDR 9.0					
Wet Return	EDR 20.0		182	53		
Risers and Branches (Bare)	66.0	1.82	389			
Boiler Shell		4.50	800	275		
Raising W.L. 2"				51		

Heating surface in square feet: Radiators 543.1, pipes 190, radiators and pipes 733, boiler shell (3% of 733 =) 22, total or boiler tax 755, piping tax (190 ÷ 543 =) 35%. Note, 22% if mains had been covered.

The domestic hot water supply load is a small quantity occurring only in Cases 1 and 2 as noted in Table 2. Water in a covered storage tank would no doubt still be at a temperature of no less than 100F by morning and would not be over 130F by the time pick-up is completed. With a 50 gallon tank this would give a tax of 52 sq ft or 57 sq ft with losses from pipe connections included.

Volume in cubic feet: Radiators 3.31, mains 4.70, risers and branches 1.82, piping 6.52, boiler steam space 4.5, total 14.33.

Weight in pounds: Radiators 1283, pipes 1245, boiler shell 800, copper 320, cast iron 963, steel 2045; water in wet return 53, water in boiler 275, but if water line is raised 2-in. add 51, total weight of water 328 or 379.

Specific heats: Copper .093, steel .118, cast iron .119.

Note 18 radiator air valves and 3 main vent valves.

Heat of refractories neglected throughout.

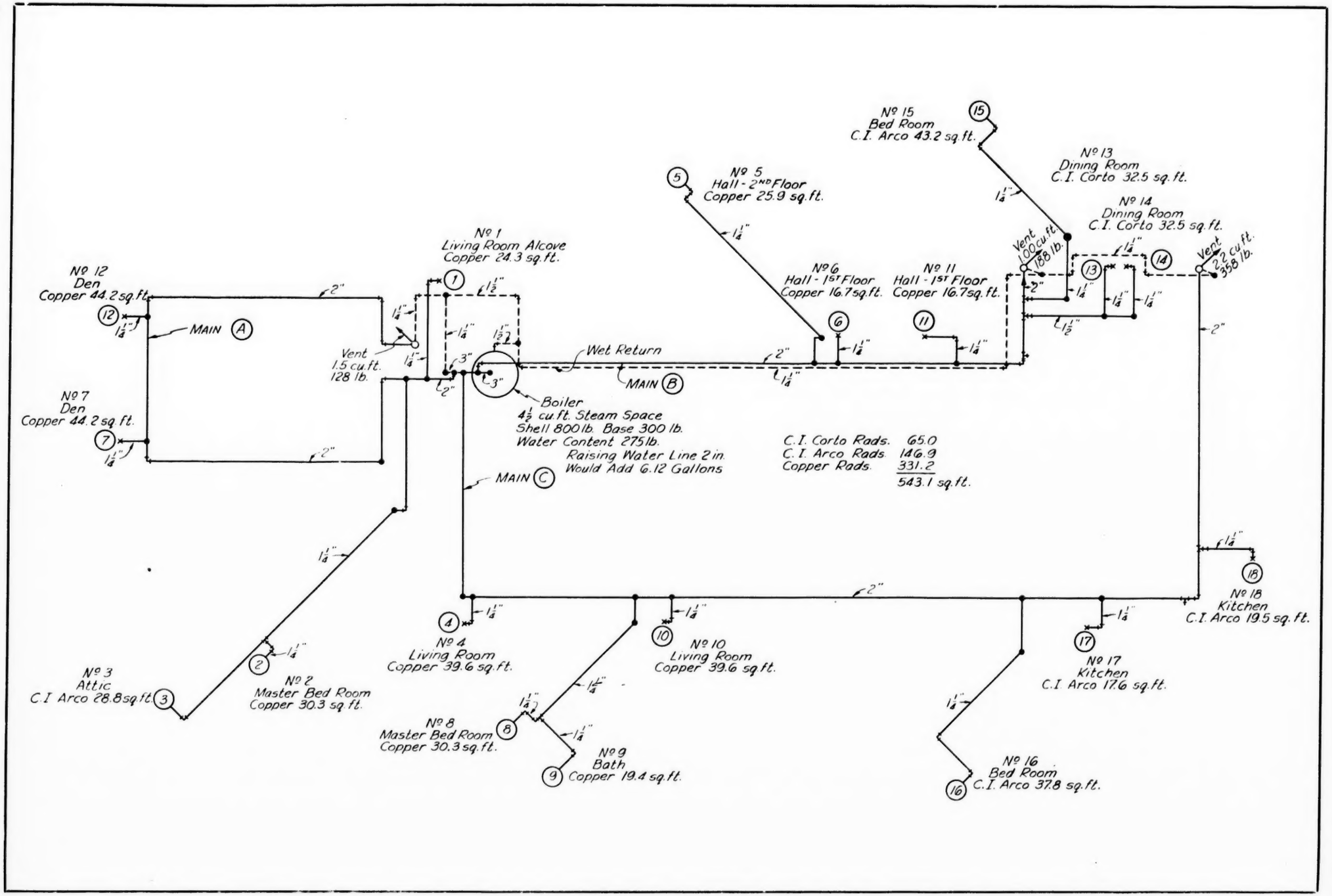


Diagram of heating system showing type and surface of radiation

air in their respective mains, one at 1.0, one at 1.5 and one at 2.2 cu ft. Piping in risers and branches contain 1.8 cu ft of air and the radiators 3.3 cu ft, making a total of 14.3 cu ft of air to be vented through 18 radiator air valves and three main vent valves when the system is not under vacuum.

Some of these figures would vary had the system been simplified by using the usual two instead of three mains and had the conditions been made more idealistic for the purpose in mind by the installation of like radiation, such as copper throughout. With such treatment there would be only 0.7 cu ft of air in the radiators, a substantial reduction from the 3.3 cu ft, but the air in the system would not be materially reduced. The relatively large volume of air contained with the boiler and piping would still remain, therefore reduction of radiator air volume by the use of like radiation does not matter, except where the fire starts with a slight amount of steam left in the system, say enough to fill the boiler and the mains.

Table 2—Pick-Up Load

(In Btu to raise temperature of metal of radiators, steel boiler shell and pipes and of water in boiler and wet return to 215F with space at temperatures noted.)

Case No.	Btu Required	System Starting Temperature, F	Space Starting Temperature, F	Remarks
*1	139,751	50	50	Hot water supply load included for Cases 1 and 2.
2	117,230	70	70	
*3	61,954	140	50	No hot water supply load in Cases 3, 4, 5, and 6.
4	53,527	140	70	
*5	40,543	170	50	
6	32,116	170	70	

*The additional heat given off by all metal (radiators, pipes, and boiler shell) when surrounded by air at a temperature of 50F is $\frac{240 \text{ Btu}}{215-50} \times 4\% \times (215 - 50) \times 755 =$ 7248 Btu. This amount is included in Cases 1, 3 and 5. The coefficient of heat transmission varies 2% for each 10F variation from standard temperature conditions of 215F —70F.

Note, heat transmission of 240 Btu per sq ft of surface assumed for pipes, same as for radiators, as most piping is at ceiling and in hotter air.

In Cases 3, 4, 5 and 6 the aquastat is set for boiler water temperatures as noted.

Operating Load

The operating load, in Btu per hour for the entire system with space at a temperature of 70F and steam at 215F, will be

755 sq ft $\times$ 240 Btu $=$ 181,200 Btu per hr.

The system, however, will supply this heat loss in less than an hour since the surfaces are at a temperature of 228F corresponding to a pressure stat setting of 5 lb. The transmission will then be at the rate of 268 Btu per sq ft per hr, hence the hourly heat losses will be met by the surfaces in

60 minutes $\div$ [268 $\div$ 240] $=$ 54 minutes.

As for the burner, it may run only 40 minutes out of the hour to supply the load (both normal and pick-up) where the aquastat is set at 140F. During the pick-up period the surfaces will transmit some heat to the space and in the cooling off period will give up some of their heat to the space.

Time Required for Pick-Up

Assuming that the burner can supply the normal load of 181,200 Btu in 40 minutes, or at the rate of 4,530 Btu per min, then the time required for the burner to supply heat equivalent to that required for the pick-up would be as follows:

Case 1—32	min. or 44	min. per corrections* noted below
Case 2—26	36	
Case 3—13½	18½	
Case 4—12	16½	
Case 5—9	12	
Case 6—7	9½	

*Corrections. During the pick-up period, the surfaces will be transmitting heat to the space, which amount for a zero day outdoors should be an average of one-half of 181,200 Btu in 54 minutes, or 1,685 Btu per min. The increase in time and load for each of the foregoing cases would be 1685 $\div$ 4530 $=$ 37%.

The necessary corrections are as appear in the previous table and represent the time required to bring everything up to normal and meanwhile supply the heat transmission from the surfaces at their lower transmitting values.

Time Required to Cool Off*

Since the surfaces have an average transmission in cooling down of 1,685 Btu per sq ft per min, the time taken to cool off will be as follows:

Case 1—86 minutes	Case 4—32 minutes
Case 2—70	Case 5—24
Case 3—36	Case 6—19

For Cases 1, 3 and 5, it is assumed that the house will cool down to 50F in the same time as given in the cases cited. If the heat losses from the house lag behind the heat transmission, the time will be slightly less according to the case and the temperature of the space.

Operation After Burner Shuts Off

As the burner shuts off, the steam pressure within the system quickly lowers to a partial vacuum. The steam remaining quickly condenses and stops circulating unless the air valves are of the vacuum type, as there is no pressure to keep out the air. Even with vacuum air valves, as in the present case, circulation

*No doubt the given figures under these sub-headings are theoretically incorrect since the transmission in cooling down is not necessarily a straight line. It may be very rapid at first, then very slow over a long period. Perhaps it cannot be figured. The figures are given, however, because they seem to agree fairly close with the conditions in the house in question. The comments of anyone sufficiently interested in this sort of problem will be welcomed.

of steam is brief. There just isn't enough sensible heat stored in the water and metal of the boiler and wet return and in the burner refractories to provide the latent heat of evaporation to circulate the steam required unless it is a coal fired system with a banked fire. The time the system will continue to circulate may be figured as follows:

The stored heat from 228F to 170F is 25,810 Btu and the average transmission load is 1,685 Btu per min, hence in (25,810 ÷ 1,685 =) 15 minutes the boiler water will cool to a point where steam will no longer circulate. In 23 minutes it would cool to 140F, the usual aquastat setting, a temperature corresponding to a vacuum which would be difficult to reach and much more so to maintain. With a high aquastat setting and a very tight system, it is possible for the burner to come on for hot water at the aquastat setting as or before the stored heat is all used up. In such an event, circulation of steam could still be maintained under the vacuum.

The latent heat of steam at 18 in. of vacuum (170F) is 1,133 Btu, the volume per pound of the steam is 63.4 cu ft and the volume within the system, exclusive of the steam space is 9.8 cu ft, hence the steam would remain in the radiators and the system until about the last six or seven seconds of the 15-minute period.

Raising the Boiler Water Line

Raising the water line 2 in. in the boiler would not increase the heat absorption of the boiler in question, but would increase the amount of stored heat above 170F by 2,958 Btu, enough to lengthen the 15-minute interval to 16.7 minutes, not a gain of any consequence. The effect on the pick-up period would be to lengthen the period by from ½ to 2 minutes from Case 6 down to Case 1. The additional time interval increases the duration of loss due to stack temperature. The higher water line would provide a safety factor against low water, but might produce carry over of water. A boiler with a large water content could be used to obtain the same condition as with a normal water line.

The burner in this particular system can carry the normal load in 40 minutes. It is large enough to carry the usual pick-up load (Case 4). Only for the unusual cases (Case 1 and 2) should the oil burner nozzle be made larger in capacity, about 20% in these two cases, or else the burner can be run longer.

To meet the conditions (Case 1 and 2) an oil burner capacity of 3½ gal would be required, assuming oil at 141,000 Btu and 70% efficiency. A capacity of 2¾ gal would be sufficient were it not for these two cases. The installation, however, has a burner nozzle of 2½ gal capacity, having been reduced from the original of 3 gal in order to save fuel and also to reduce the amount of combustion noise.

There is a nice balance between a larger nozzle giving a larger fire and therefore a higher stack temperature and a smaller nozzle with a lower stack temperature of longer duration, but with less standby loss. A 3-gal nozzle would have reduced the running time of 40 minutes to 33 minutes and the various pick-up loads by 17%. It, however, made the burner operation more frequent. Where the burner cuts off by pressure stat before the room temperature is satisfied, it shows that the heat absorption characteristics of the house lag behind the power of the boiler and burner. If these two things were in balance the burner would cut out by pressure stat and room thermostat at the same time.

What Happens with Less Radiators Turned On

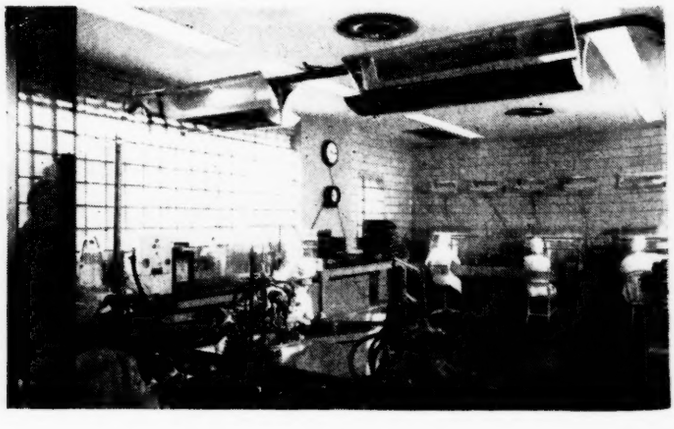
Suppose that a few of the radiators were valved off, so that the heat transmission losses to the house would not exceed the heat losses from the house. In such a case, the radiators would be hot and the burner on for proportionately more minutes of each hour, but the burner operation would be more frequent due to the pressure stat cutting the burner out more often.

Air Conditioned Pharmaceutical Plant

In the Winthrop Chemical Company's new controlled-conditions Ampul building at Rensselaer, New York, production of more than 200 highly-developed products is carried on under dust-free, germ-proof conditions. The structure has walls of face brick with glass block panels and limestone trim. Insulation and vapor barriers in the walls, and shade screens outside the glass block help in air-conditioning the building, which is "sealed in" completely.

Outside air is taken into the conditioning system through louvered openings and throwaway type filters and passes through precipitrons and over germicidal lamps as it enters the conditioning system. Used air is exhausted through louvers in the masonry on the front and rear walls. Building was designed and erected by The Austin Co.

A heavy concentration of germicidal lamps in the Niphanoid and Novocain room illustrated, insures purity of these products. This area, which is typical of the 35 isolated laboratories in the new building, is subject to special zone controls which maintain 25% relative humidity, with a maximum deviation of plus or minus 2%.



Needless Regulations Impede Use of Arc Welding in Industry

A. F. DAVIS

Vice President, The Lincoln Electric Co., Cleveland, Ohio.

Regulations, superfluous in the opinion of the author, are hindering wider use of arc welding. He points to inconsistencies in existing regulations and to the fact that in many cases, the weld is stronger than the material being welded.

WHILE codes establishing specifications for the construction of products where safety is a salient factor are vitally necessary, many of the current regulations and interpretations resulting from these codes are hindering rather than benefiting the advantages inherent in electric arc welding.

Such needless regulations create in the minds of potential users the feeling that arc welding is veiled in a cloud of doubt and suspicion, and that it is only because of these codes that welding is a proved application in industry today.

We can well see that every precaution must be taken when a locomotive boiler is being welded. But, on the other hand, there are innumerable other applications of arc welding where the excessive requirements of the codes create higher costs, greater time and needless concern regarding its application.

The regulation, for instance, that requires that *all* welds under code requirements be X-rayed falls under the superfluous category. This test, the cost of which is always added to the cost of welding, increases unfairly the cost of the welding process in any given operation. This is unfair because so many of the applications do not involve human safety. The welded joint has been proved under all circumstances to be stronger than the parent metal and is tremendously stronger than any riveted joint where the use of X-ray has never been suggested.

This is best illustrated by Fig. 1. The specimen contained a welded joint that had abundant surface porosity which is not permitted under the code. During this tensile test, the bar failed not at the welded joint but in the parent metal instead. This proves that a weld, classified as faulty by the existing code, is more durable even when faulty, than the structure of an ordinary bar.

Still these regulations prevail on the testing and minute inspection of welds. Small wonder some new

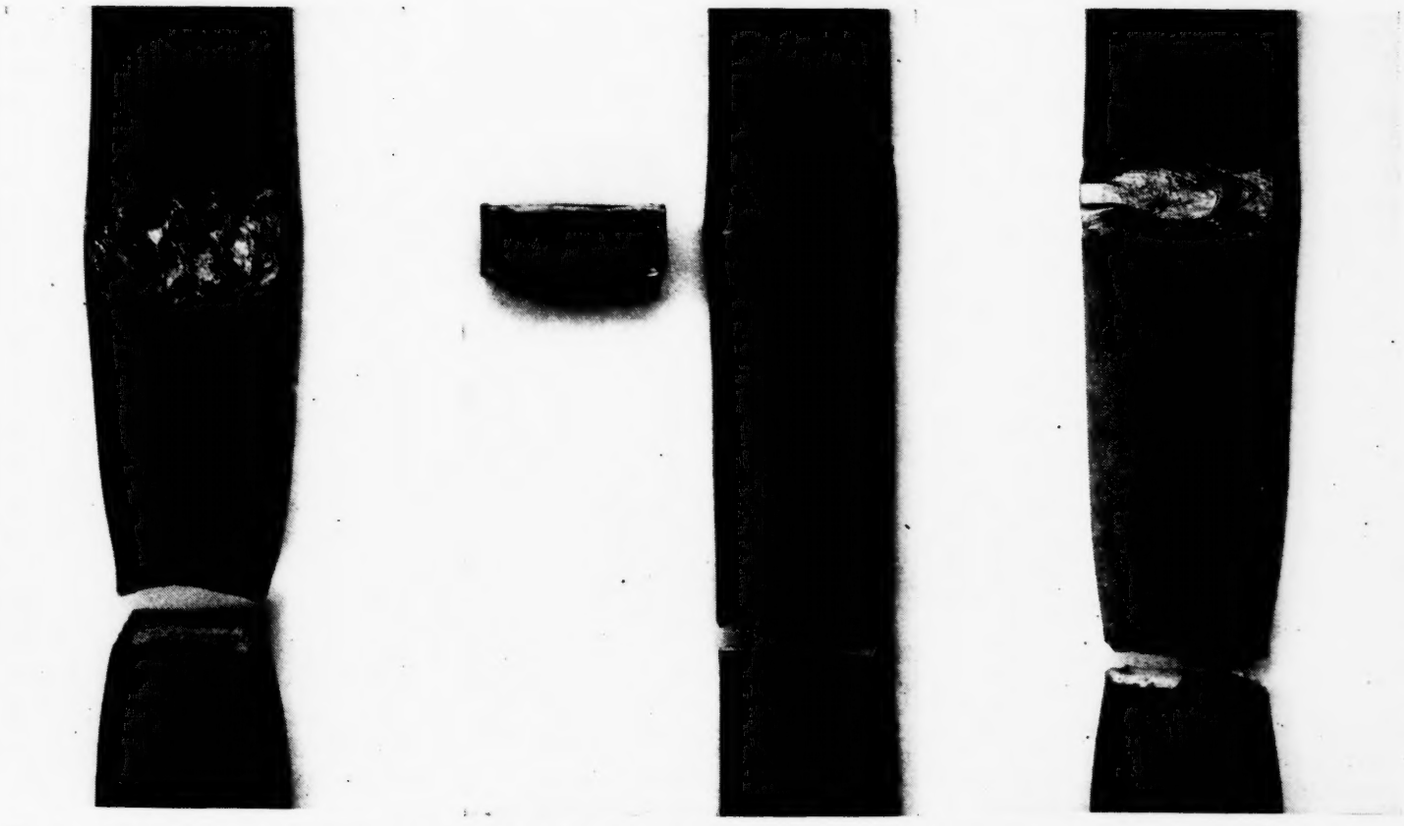


Fig. 1. (Left) Tensile specimen failed at 59,000 lb per sq in. Welded joint had abundant surface porosity but failure occurred in the plate itself. Fig. 2. (Center) Weld was rejected because of internal porosity in the weld. Parent plate, not weld of tensile specimen, failed at 62,500 lb per sq in. Fig. 3 (Right) Weld with an abnormal amount of undercut. Specimen failed in the plate at 59,400 lb per sq in.

users of the arc welding process get an erroneous impression and feel that perhaps there are some grounds for suspicion in the case of arc welding versus riveted and bolted joint structures.

Welded joints are chipped out and rewelded, rejected because of trivial and infinitesimal defects existing somewhere on the surface or beneath, yet the parent metal is never as perfect in physical structure as is the arc weld properly made. This is illustrated by Fig. 2, where a tensile specimen with a welded joint which had an abnormally large amount of internal porosity, was pulled to failure. The failure was not in the weld but in the parent metal of the bar.

Riveted joints are frequently made tight by caulking. This method is accepted without any questions asked. The undercutting is enormous and far deeper than a slight undercut on a weld that is rejected for this very reason. But there is never a question raised in regard to the caulking of a riveted joint. But a weld, because of slight under-cutting, is rejected under the prevailing codes. Fig. 3 shows a tensile specimen containing a weld with an abnormal amount of undercut. Here again, under test, the specimen failed not in the weld but in the parent structure of the plate.

The riveted joint, as we all well know, is invariably the weakest spot on any member or structure. But this condition never exists in the case of a weld, for it is always the strongest spot of any member or structure.

Electrodes are manufactured under strict specifications. Rivets are never manufactured under any code or regulation that specifies they have certain positive qualities in regards to tests.

Contour and ductility of welds are salient points from the inspector's standpoint, but this same inspector never once checks the shape of the rivet nor the hole that it only partially fills.

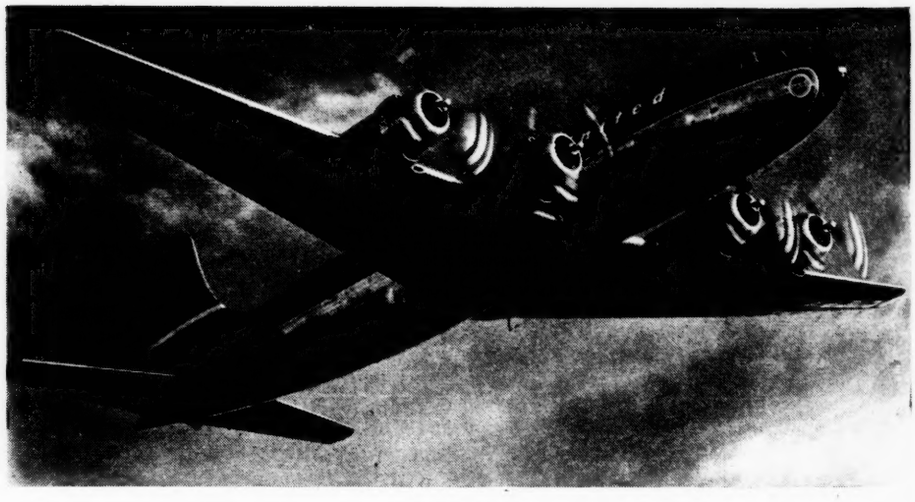
Examination of welders' qualification often is unfairly conducted. Some welders fail to qualify for a second time, after they had successfully passed the requirements the first time. This reflects not necessarily on the operator's ability but, to a large extent, on the quality of the parent metal that the operator is welding. This metal may have been purchased to the same specifications as that previously used and on which no difficulty was encountered.

Operators almost invariably encounter their failures on the guided root bend tests. Examination of the failure usually shows a small spot where fusion was incomplete. This could be considered a satisfactory explanation for the failure, except that often samples which have obviously greater imperfections will bend without a sign of failure.

Both of the samples are prepared in the same way, welded with the same electrode and by the same qualifying operator. But the qualities of the base metal differed in the two tests.

The use of high tensile strength electrodes on the higher tensile strength plate will normally not be of much help. It will not improve the ability of the plate to bend. Therefore, it is only right to say that the guided bend test does not always provide a fair test for the qualification of operators. As the test now stands, it is possible to pass a poor operator by having him qualify on good pipe or to fail a good operator by having him qualify on poor pipe.

Air Conditioned Flight



Aviation heating and ventilating engineers have determined that, during the winter, interior temperatures of airliners such as United Air Lines' new Mainliner 300 (Douglas DC-6) should range from 68 to 72F. In summer, most comfortable temperatures are between 74 and 78F. Travelers are dressed more warmly in winter, they say—hence, lower cabin temperatures for that season.

United's five-mile-a-minute Mainliner 300 is designed to provide air travelers with living room comfort, regardless of outside air temperatures, while cruising at 15,000 to 20,000 feet. Cabin pressurization and temperature conditioning are accomplished through a supply of air from two engine-driven superchargers located in the outboard engine nacelles. The heating system incorporates an internal combustion heater.

An air-cycle refrigerating unit, consisting of an inner cooler, expansion turbine and axial flow fan, operates in parallel with the heating system. The ventilating system can deliver conditioned air to the cabin at a rate of approximately 20 cfm per passenger. Cold air ducts are accessible to each outside seat, adjustable by the occupant.

Minor Adjustment Changes End School Room Complaints

JOHN CASSIE

Research Engineer, B. F. Sturtevant, Division of Westinghouse Electric Corp., Hyde Park, Mass.

Minor adjustments to fan equipment eliminated the cause of complaints regarding heating and ventilating of schoolrooms.

ARE you detective-minded? Here is an interesting account of how starting from a series of complaints regarding heating and ventilation in schoolrooms, it was possible to locate the source of the trouble, and by means of minor adjustments eliminate the difficulties. This account is related because it may help others faced with similar cases, solve their problems.

This heating and ventilating system was installed in a modern, combined senior and junior high school. The fact that it included the latest type woodshop, machine shop, forge shop, and domestic science rooms was of great help, since most of the corrective alterations to the system were of necessity accomplished at times when such classes were not in session, particularly over the weekend.

Many complaints were directed against the allegedly unsatisfactory pieces of equipment incorporated in the heating and ventilating system. The fan equipment bore the brunt of the disparaging remarks.

1. The fans, running at about specified speeds, were handling, during most of the field tests by the customer, only about 50% of specified volumes. It was claimed that these low volumes, combined with brake horsepowers exceeding those specified, precluded the possibility that the lack of capacity was due to excess system pressure losses. Conclusions drawn were that the fans were deficient in capacity and efficiency.
2. Complaints of a "terrible draft from the supply grille" and that she "Just couldn't get the room warm in spite of the thermostat setting" were made by one English teacher.
3. From the room across the corridor from the English classroom came an opposite complaint—"too much heat."
4. In certain rooms arose complaints of "noisy fans."
5. Some supply grilles emitted puffy air streams.
6. At times, some classroom doors were difficult to open. The same doors caused no complaints during the balance of the day.

Since the capacity and efficiency of the fans must be corrected, regardless of whatever other troubles existed, this was attempted first.

Each fan was installed in an equipment room of ample size, drawing its air from a true plenum chamber from which the air flowed to the fan wheel inlet without prespin and in such a manner that each blade of the wheel received its air in the designed direction.

Thus, one of the possible causes of abnormal fan performance was eliminated. The wheel checked in design details and proper rotation. In some instances the axial relation of housing inlet piece and wheel inlet was incorrect.

Investigation revealed that in an attempt to line up fan sheave with motor sheave in the field, the fan rotor (including wheel, fan shaft and fan sheave) was translated axially. Therefore, the proper relation of wheel to casing at time of shop assembly, was upset. Complaint No. 5, puffy air-streams, was eliminated by reestablishing the correct relation of fan parts.

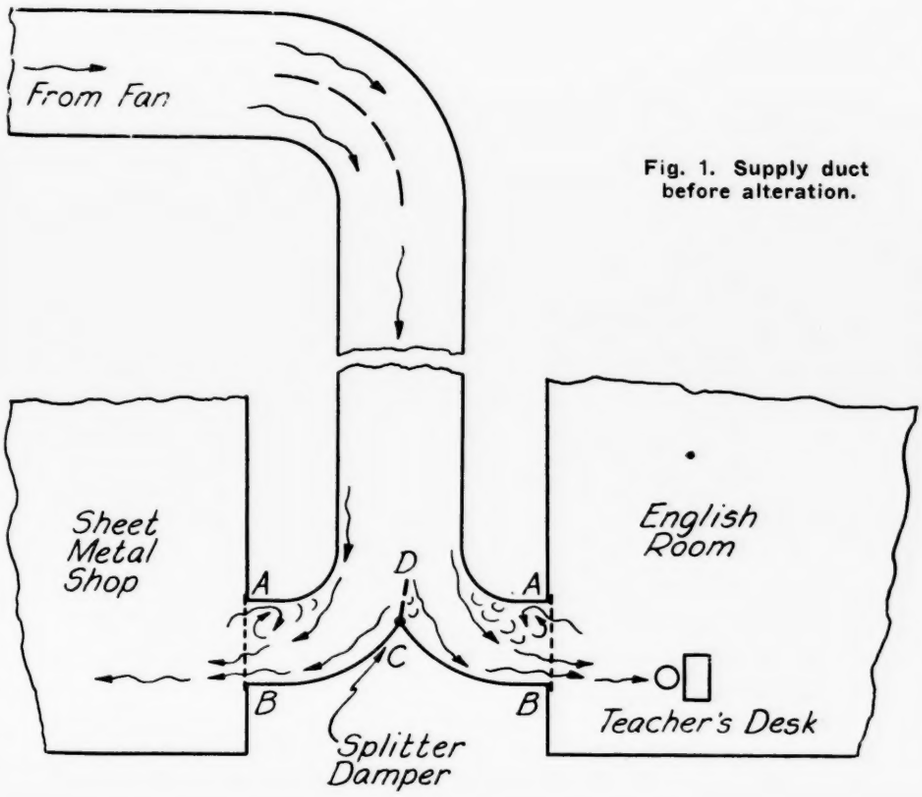


Fig. 1. Supply duct before alteration.

While making the above adjustment some loose bolts and screws were discovered. Tightening these bolts and otherwise preventing steel plate sheets from vibrating eliminated noises which once created had carried throughout the duct system to the serviced areas. This cancelled complaint No. 4 regarding noisy fans.

As the fans were now absolved from complaints of a mechanical nature, investigation of the installed air performances was continued.

The typical fan discharge duct arrangement is shown in Fig. 1. The duct, installed just above the corridor ceiling, discharges from its branch ducts through classroom wall grilles into the connected classrooms at a level just above the door tops. Air is exhausted from the classrooms through louvers in the door panels near the floorline to the corridors which are used as return air passages.

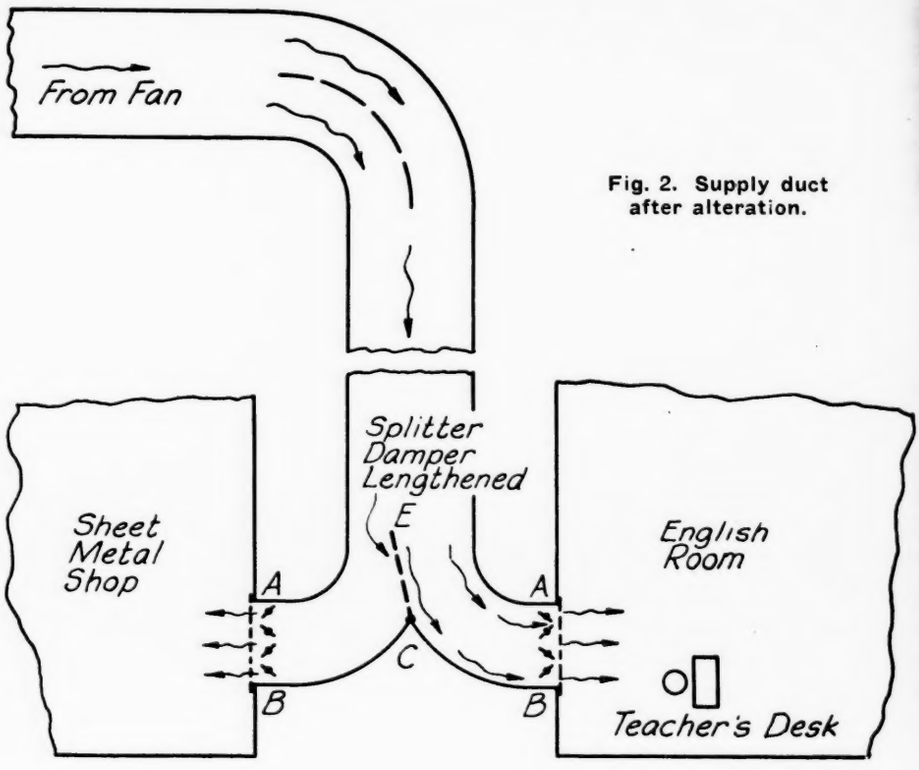


Fig. 2. Supply duct after alteration.

Due to the nature of the elbows *AB*, the streams crowded toward outside radii at *B*; near *A*, the indicated eddy was set up, causing reversed direction of air flow as shown. The excessively high velocity at *B* was sufficient to form a jet across the English teacher's neck—hence, her complaint of drafts. Splitter damper *CD* was found to be too short to be effective.

In the sheet metal shop, the occupants were too warm due to the nature of the classroom work.

Complaints from both rooms were eliminated when the system was altered as shown in Fig. 2. In this revised system, the splitter damper was lengthened to *E*. The damper position was set to favor the English classroom, while decreasing the supply to the sheet metal shop.

Jetting of air into the room was eliminated by installing and adjusting "squeeze dampers" at *AB*. These dampers were made from boxwood in the school's woodshop. The wooden damper leaves were pivoted on finish nails. Similar dampers fabricated from steel plate, are obtainable. These corrections eliminated complaints 2 and 3.

These systems were intended to be operated with classroom doors closed, and exhaust air passing through door panels to corridors. While measuring the air volume supplied to the English room with its door closed, it was discovered that the door of the sheet metal shop, and those of other rooms connected to the same system, remained open part of the time and closed the balance of the time.

This meant that, during the volume determination test in the closed-door English room, the system fan was supplying an over amount of air through the other rooms by way of their open doors. The fan was not given credit for the excess capacity which flowed through the open-door rooms since, when these rooms were under test, these doors were closed and the English room might then have had its door open, sending through more than the credited air volume.

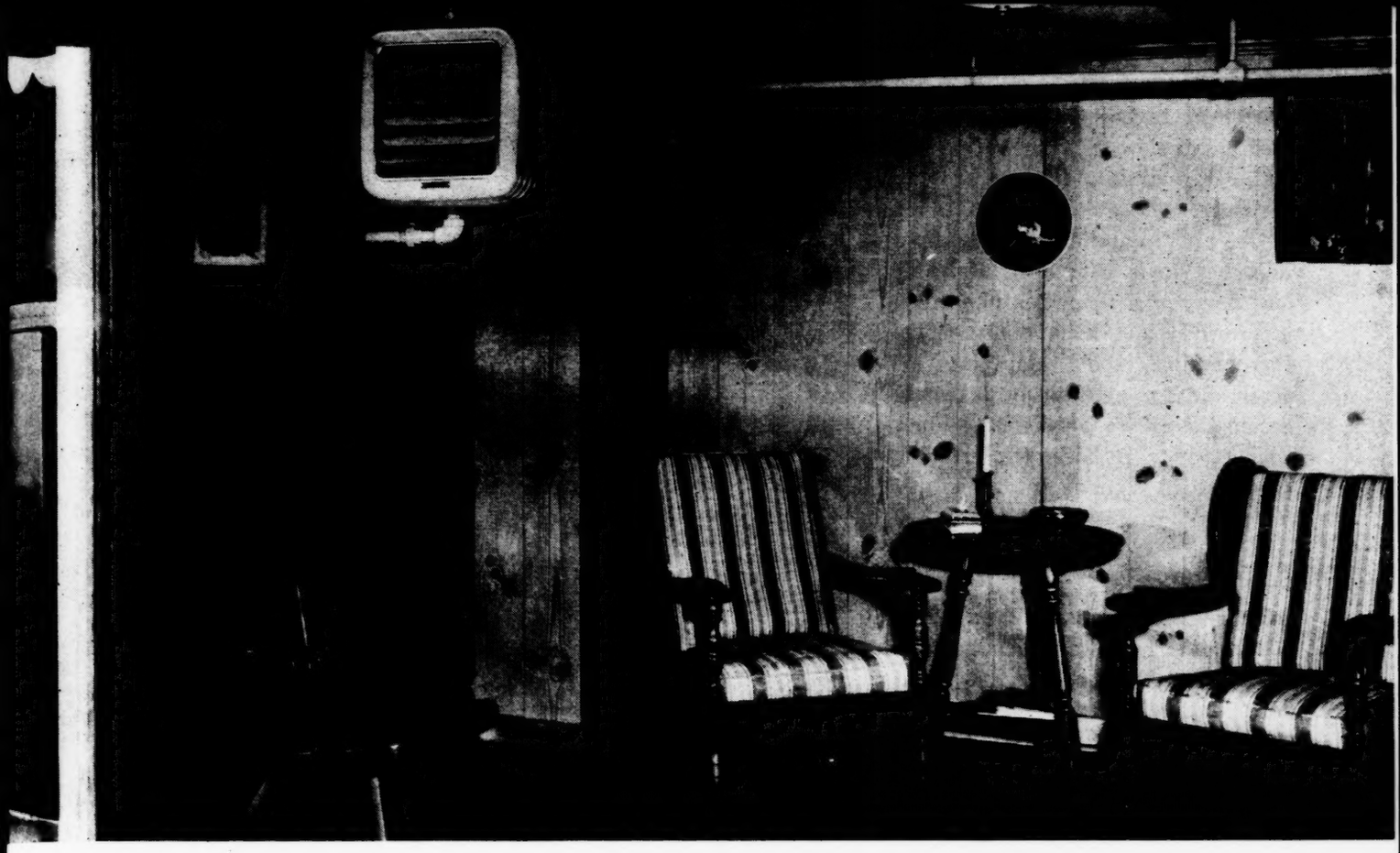
Human nature being a wide-range variable, it was not feasible to attempt forcing the personnel to keep all doors closed. To get control of the situation, tests were run at night while all doors were locked. The night janitor had the only duplicate key and he cooperated by closing each door immediately after using it. Under these controlled conditions there was no difficulty in convincing the customer that fan catalog ratings were satisfactorily met.

With all doors closed, the air actually whistled through the door panel louvers to the corridor. Tests confirmed excessive pressure drop through the louvers and a correspondingly excessive velocity. It was arranged that louver areas be increased. These changes relieved the pressure within the room, reduced the resistance to be overcome by the fan, and thus allowed a larger volume of air to flow through the system at a given fan speed.

Where volumes increased too much, sheave changes for speed reduction were recommended. Since the corresponding horsepower decreases as the cube of the speed decrease ratio, the required capacities were attained with greatly reduced horsepowers, due to elimination of louver resistances. Speed reduction also decreases the air flow noise for a given volume handled.

Decrease of louver resistance eliminated complaint No. 6—difficulty in operating classroom doors. Whether doors are now left open or closed makes little difference in volume of air handled since the louver resistance is now negligible.

Thus, *with the original fans and coils in place* the system is now considered satisfactory.



Unit heater in basement recreation room gives quick service for impromptu parties.

Unit Heaters Ideal for Home Basements

M. J. DRUSE

Modine Manufacturing Co., Racine, Wis.

WHILE unit heaters are primarily designed for industrial and commercial space heating, there is nothing in the rule book to prevent their being used to excellent advantage in home basements. In fact, in recent years, an increasing number of home owners have discovered for themselves, the unique features of basement unit heating. This may have been brought about by the transition of basements from boiler rooms to attractive and useful recreation and hobby areas.

However, no amount of pine panelling, asphalt tile flooring, modern lighting or clever furnishings will assure continued use of the basement if the rooms "down under" are cold, damp and musty. Furthermore, such conditions promote the rapid deterioration of originally attractive furnishings.

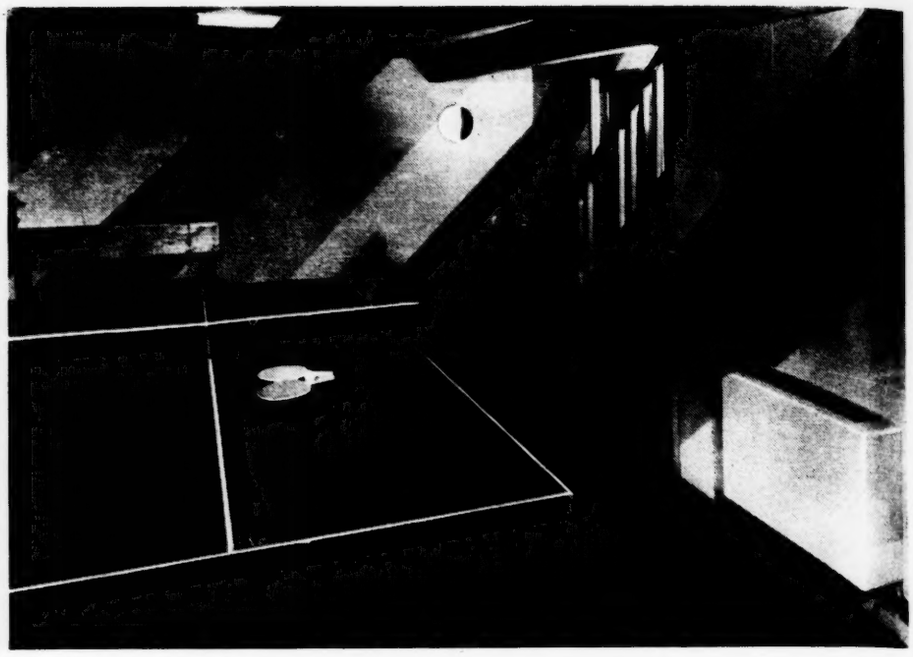
In the average sized recreation room, one small unit heater connected to the boiler, will usually solve this problem. Operated manually or thermostatically, it will supply the required heat and air circulation to maintain perfect comfort, eliminate cold floors and counter dampness and mustiness. In other words, it will bring "upstairs comfort" downstairs. If properly sized and located, the same unit will serve adjoining rooms. In this connection, the use of louvered doors permits circulation of heated air throughout the basement.

With unit heaters, there is no waiting for warm-up. When guests arrive unexpectedly . . . intent on a game of ping-pong, the recreation room can be made comfortably warm in a few minutes by operating the unit

Same heater helps dry clothes on rainy days.

heater at high speed. Thereafter, medium or slow-speed operation will suffice to maintain the proper temperature.

The air circulation provided by a unit heater is especially desirable on "party nights" when the room is crowded and filled with smoke. Introduction and tempering of fresh air for ventilating is easily accomplished by means of a duct connected to a basement window from which a pane has been removed.



Wall unit helps keep cellar dry and livable.

Summer Benefits

Year 'round operation of the boiler for heating domestic hot water, permits the unit heater to be operated during the summer months . . . just often enough to keep the basement dry and fresh. Even if hot water or steam is not available, circulation of basement air by the unit heater fan is helpful in preventing a musty, dank condition. And to go one step farther, it is a simple matter to recirculate basement air over a calcium chloride dehumidifier of the type sold in all department stores . . . Thus the recreation room becomes livable the year around.

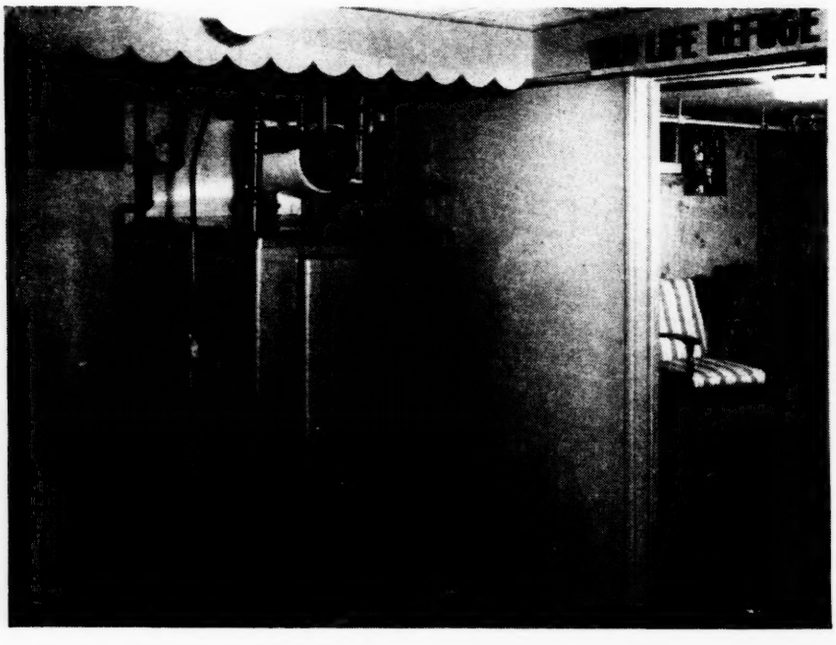
For Drying the Wash

Cold and rainy wash days are of little concern to the housewife having access to a unit heater-equipped basement. She has at her disposal the same clothes-dryer used by many launderies and cleaning and dyeing establishments. Operated at high speed, the unit that heats the recreation room, will also provide the rapid circulation of heated air required to dry the day's washing in a jiffy.

Converting the rumpus room to an efficient drying room, involves only a wash line and hooks . . . or one of the patented collapsible laundry racks now on the market. Wash to be dried should be arranged to permit unrestricted circulation of heated air around it . . . preferably the air passing between the rows of wash. Normally, a wash line which can be quickly put up and taken down without much labor and trouble will be found suitable.

When used for drying, the unit heater may be operated first, with the heating medium circulating through it, then for the finishing stages, with just the fan recirculating the air.

And speaking of drying . . . those who have unit heaters in their basements invariably "discover" them to be the last word in hair drying equipment. Although a far cry from its primary application of factory heating . . . it works ideally for that purpose.



Types of Unit Heaters Applicable

For basement use, the propeller fan, horizontal discharge type, or the blower fan, cabinet type are usually recommended. If hot water is to be used as the heating medium, care should be taken in selecting a unit heater having a relatively high ratio of fins to tubes, assuring a comfortable final or delivered air temperature. While forced circulation is usually recommended for hot water application, there are many successful gravity jobs in use. The latter is usually permissible when runout from boiler to unit is short.

(Left) Automatic heat for summer hot water gives year 'round protection against dampness.

Experience is my Classroom

By T. W. REYNOLDS

PICK-UP LOAD

The pick-up load is made up of the following items: additional heat required to raise the temperature of the circulating medium from lowest temperature anticipated to normal circulating temperature, plus the heat for raising the temperature of metal in boiler and system to the temperature of the circulating medium, plus the excess heat output of all metal when surrounded by the decreased temperature of the surrounding air. The last item is seldom taken into account.

If the reserve output of a boiler is small, it is advisable to keep the space temperature near normal in the few days of extremely cold weather, so as to eliminate this factor during the pick-up period.

The period of maximum heat demand should also be anticipated in order that ash and clinker can be removed and a deep fuel bed of good quality coal provided which will be well ignited by the start of the pick-up period. This is more important with anthracite than with bituminous, for anthracite is slow to ignite. Drafts must be adequate or there can be no pick-up.

PIPE LENGTHS

When you see a truck load of pipe you will note a red flag hanging from the rear of the truck and fastened to a length of one of the pipes. No matter how small, soiled or shop worn this rag may be, it seems to satisfy the law.

These long lengths of pipe are a hazard and a nuisance all the way from mill to truck to turns in the alley or building and won't go up the elevator. Brass pipe is made in 12 ft lengths, but it is still pipe. Such lengths would fit in a box car and the steamfitter's truck and would have many other advantages.

Note: Mr. Reynolds is Chief, Air Conditioning Division, Abbott, Merkt & Co., Engineers, New York.

AN OVERHEATED GARAGE

The garage had too much heat—an unusual complaint—so the figures were checked and found interesting. An addition was made in the calculations for exposure, yet the roof was flat and knew no cardinal points of the compass; in fact much of the time the place was being fairly well heated by sun effect on the large roof area. There were also many cars which came in cold in part, but heated inside and with motors and radiators hot.

OUTLETS AND PARTITION CHANGES

How to design the air conditioning outlets in a department store, so that they are fixed for six weeks or six years after installation is one of the toughest problems in air conditioning. Partitions are here to-day and gone to-morrow, for the desires of the customer must be gratified as to the kind of merchandise when wanted in the quantity wanted. Thus, a run upon nylons may lead to the acquisition of so many husbands that diapers become all the rage.

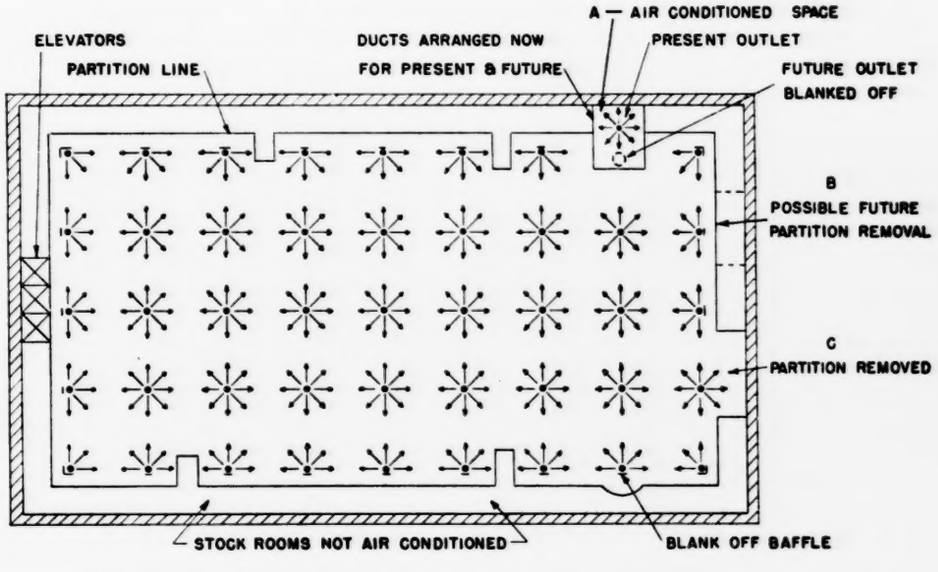
As a rule, however the space alloted to each item is proportioned in area (at least once a year) in accordance with the relative sales of that item during the past year and the value of the space so allotted. Now these partition changes may block the flow of return air and affect the throw and the path of air from supply outlets, though ceiling outlets (if made one to a bay) are usually less seriously affected than are side wall outlets.

In the modern department store with large open areas, partitions are usually located only along the exterior walls. The line of these partitions will fluctuate in and out and be subject to divisions, somewhat as typified in the accompanying sketch.

Since one does not design in a paradise and must temper his engineering with business, the speed of the project and his future with the customer, he must yield to the inevitable and do the next best thing under the handicaps as enumerated.

The ceiling outlets should therefore be so located (see sketch) that they are not so close to a partition as to be seriously affected thereby, yet near enough so that partition removal will allow sufficient coverage of the space beyond the partition. This is illustrated by *C* in the sketch where a partition has been removed and the outlet now covers a space included within the air conditioning space. This space had been a stock room and was not air conditioned, but even if it had been by means of a separate outlet, such outlet could not be left for it would neither be sized or lined up to match the others in the open space. Similarly, note at *B* the outlet which may some day serve the space now enclosed, but not conditioned.

At *A* there is a separate space which must now be air conditioned by a separate outlet. This space might be changed back to a stock room some day. If and when this is done, the separate outlet can be removed, its



Air conditioning ceiling outlets in upper floor of department store.

connection from the duct blanked off and the ceiling opening closed. A new hole in the ceiling can be cut at the outlet marked "future." Through this opening a new outlet can be inserted and connected to the duct as now blanked off.

All this avoids a minimum of changes and damage to ceilings. Ducts, of course must be piped for present and future, and outlets must be readjusted for air delivery as changes are made. Selecting outlets for the open area all of one size, yet with sufficient range between maximum and minimum air quantities allowable by that size will permit readjustments up or down of the outlets air capacities, yet without causing drafts from the outlets.

HOT PARTITIONS

When you are calculating the individual heat gains for a multiplicity of rooms (for example, an office layout), the occasional hot partition, floor, or ceiling looms up as a nuisance. It alters the routine, a pause is required for thinking. Arbitrary assumptions are then quickly made as to the temperature prevailing on the unconditioned side of the partition. Such assumptions are usually on the safe side, but so much so that the air supply to the room (to meet the heat gains so computed) becomes too great for a normal air movement within the room.

Exterior corridors along air conditioned offices are often left without air conditioning, though in many cases some conditioned air should be supplied to the corridors as well. This would eliminate the hot partition and give better results within the offices, yet with no increase in refrigeration load. In fact, according to the calculations of many engineers there would also be a substantial reduction in the total air requirements for the fan.

Consider the various conditions met with when calculating the heat gains from hot partitions, somewhat as shown in the sketch. For the sake of simplicity, assume that conditions on the floors above and below are also as shown. Accept also the coefficients of heat gain as 0.39 for the usual partition and 0.25 for the usual exterior wall. For convenience, the rooms may be assumed as square and the typical office shown may be considered as representative of a number along the corridor.

It is customary to assume an unconditioned space as about 5F higher than outdoors. This would give a temperature of 100F in the corridor and require too much air supply in the offices, because the hot partition for the given conditions has a rate of heat transfer of 56% over that of the exterior wall. Obviously, the heat gain from lights in the corridor must equal the cooling effect of the partitions or the temperature in the corridor will drop below 100F, or for that matter even below the outdoor temperature of 95F.

The real temperature (X) in the corridor may be calculated as follows:

$$0.25\ (X-95) = 0.39\ (80-X)$$

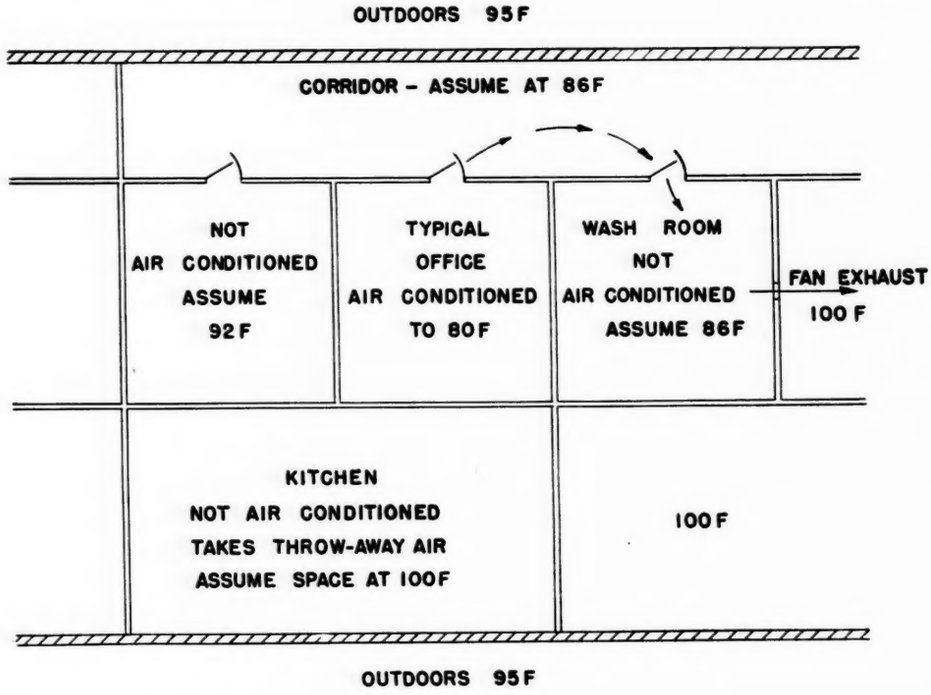
$$X = 86F$$

If the offices were conditioned for 78F rather than 80F, X would become about 84F, and even if the coefficient for the partition were doubled X would only change to 84.5F.

This shows that it is usually safe to assume corridors at a temperature which is an average of the conditioned space and that outdoors, especially so as the slight heat gain from lights in the corridor was not taken into account in the prior calculations. If throw-away air from the offices goes to the wash room by way of the corridor, the temperature in the corridor may be lowered to an extent depending upon the quantity of throw-away air and the internal heat gain of the corridor.

The temperature of the unconditioned room may be taken as an average of the surrounding spaces at 92F, or it may be calculated, in which case it will still be 92F. The wash room has conditions identical with this room, but it has the advantage of throw away air. Its temperature, considering the usual slight heat gain of such spaces, may be taken the same as for the corridor.

A kitchen as a rule receives a lot of throw-away air and may be estimated as having a temperature of only 5F higher than that of outdoors. Partitions between the conditioned spaces and the kitchen may be insulated, but the cooling effect of these partitions is of help in alleviating temperature conditions within the kitchen.


OUTDOORS 95F
CORRIDOR – ASSUME AT 86F
NOT AIR CONDITIONED ASSUME 92F
TYPICAL OFFICE AIR CONDITIONED TO 80F
WASH ROOM NOT AIR CONDITIONED ASSUME 86F
FAN EXHAUST 100F
KITCHEN NOT AIR CONDITIONED TAKES THROW-AWAY AIR ASSUME SPACE AT 100F
100F
OUTDOORS 95F

A combination of various hot partitions as encountered in air conditioning.

SKYSCRAPER RADIATOR TROUBLE

Many years ago a new type of radiator was installed in a skyscraper office building in New York. Water hammer developed in the radiators of the building, particularly on Monday mornings. The noise only lasted a fraction of a minute after the valve on a radiator was opened, the time varying with radiator size. The trouble was attributed to the quality of street steam supplied to the building, but the steam was found to be dry. The trouble was then blamed on the quickness of the radiator valve, the slowness of the radiator trap, and to most anything else that came to mind.

A few things were soon evident. The noise within a radiator could be stopped by throttling down its radiator valve. The radiator was of a type having comparatively little volumetric displacement for a given amount of condensate, and the enormous steam demand of cold metal and surroundings after a week-end could be met by an instantaneous supply backed up by a pressure reducing valve from a power house. An ordinary boiler installation could not produce steam that fast, but would take twenty to thirty minutes as a pick-up period.

Perhaps it was the troublesome water within the radiator that brought water to mind as a means of finding the solution to the difficulty. Therefore the radiators were wetted externally. As a radiator dried it was noted that the first section on the supply end dried last and as this section finally dried the noise stopped. Obviously, steam in considerable quantity and pressure was flowing quickly

across the top nipple ports to pressurize all of the sections except the first, which was under vacuum. Condensate was then drawn back into the vacuum section, but as steam finally reached and filled this section to dispel the vacuum, the condensate was released and the water hammer instantly stopped.

The solution was found by fitting sheet copper orifices to the union end of each valve, the size of orifice varying with the radiator size. This slowed down the steam supply during the abnormal heating up period.

As for new installations of like nature, the situation was met by an alteration of the usual radiator bushing. This bushing, as patented by the writer, had a number of small openings around the circumference of an end which was extended so as to protrude within the radiator, thus injecting much of the steam down within the first section. A small opening was left in the end of the bushing as to allow some small amount of steam to flow across the top nipple ports. Aside from the elimination of water hammer, the device was found to be of benefit in reducing air binding, for steam distribution became equal throughout all sections.

BOILER OUTLETS

The velocity of steam that can be passed though boiler outlets depends upon steam dome volume, vertical distance of boiler water line from outlets and whether they are top or horizontal outlets, or located at the end of the boiler. Boiler outlet connections should be full size of manufacturers outlet and should preferably be vertical to a height of two or three feet above the boiler wherever possible. These things tend to keep the steam dry, especially if there is grease and oil in the boiler. Steam velocities through the outlets should be kept down to 20 or 25 fps during full boiler output in order to keep the velocity within safe limits.

Not all outlets need necessarily be used since some boilers have the same number and size of outlets regardless of the number of boiler sections, while others may have one additional outlet only after three or more sections are added. On the other hand, all of these outlets may be required if the boiler is to be driven with less efficiency at excessive output. The installation of an oil burner will also give more rapid transfer of heat and therefore more outlets may have to be used when the boiler is oil fired.

Such outlets as are used should be on the least active sections for steam and the most active for water. Outlets should be connected into a header, particularly so on water boilers where one circuit would circulate faster than another if each were connected separately to a boiler outlet. On the other hand, a poor circuit may sometimes be helped out by direct connection to an outlet. A steam drip connection to the return should be made at the first change in direction of a steam main where low headroom necessitates a low vertical connection from boiler to main.

MORE BOILER DRAFT

Draft is not always where you want it. Sometimes it is on your neck and often it is everywhere in the boiler except within the furnace. First, you have to get it into the boiler room (this requires a sufficient inlet to the room) and then it must be made to leave by way of the furnace and nowhere else.

The size of smoke pipe or chimney required for proper draft cannot be based on the size of the smoke hood on the boiler as this outlet for smoke is often made the same size regardless of the number of boiler sections. Neither can it be based on the redness of the smoke pipe; a long boiler or a long smoke pipe with short radius turns may tire out the draft to the extent that complete combustion takes place only in the smoke pipe and makes that pipe hot.

Boiler sections not drawn up tight act as cold air checks and so do loosely fitted smoke hoods, clean out and firing doors, and smoke pipes left loose at joints and at the chimney. All joints at boiler sections requiring boiler putty or cement which cannot be reached after assembly is complete should be carefully finished as assembly progresses, for they seldom ever fill up during operation. Nuts or tie rods of sectional cast iron boilers should always be loosened about one thread after erection so as to allow for expansion of the boiler when hot. The nuts should be just a little more than hand tight when the boiler is at its operating temperature.

NATURAL (?) VENTILATION

Natural ventilation seems to incur a natural dislike on the part of most everyone. You are told that this is because one man has a better circulation of blood than the other fellow and so on. This is true to a certain extent, but it does not seem to be the real answer.

If one sits by a window on the leeward side where the heat piles up, he usually opens the window alongside of him and becomes comfortable. Hot air then leaves by way of the opened window and there is no discomfort as cold air does not come back through it. But how about the fellow by the window on the windward side? He is immediately made uncomfortable due to the increase of cold infiltrating air currents at his window. This fellow then objects and becomes a crank in the mind of the fellow on the lee side. The open window isn't near the crank, so why should he complain.

Natural Ventilation

The fellow on the windward side has enough ventilation coming in without opening his window. Those far removed want him to open it, but if he opens the window from the bottom ever so little, he freezes in the breeze as the air by-passes over the radiator in a quantity and speed too fast for mixture. If the window is opened at the top the cold air drops on someone just beyond.

For years we have been told to open a window a little at the bottom and also at the top. The air is supposed to come in cold at the bottom and go out when hot at the top, but this natural convection is upset where there are pressure differences. Now the only way to alleviate these pressure differences is to have tighter windows and to supply air by mechanical means so as to pressurize the space from the inside, at least as much as possible to do so.

CHIMNEY HOODS

The installation of a sheet metal extension or hood on the chimney top to increase the draft may lead to trouble when the extension is common to flues run separately from a fireplace, water heater and the house-heating boiler. The first two then act as check drafts on the boiler flue. Separate extensions should be made for each flue. Flue linings should also extend separately to floor or chimney base, should be sealed tightly from one another and provided with individual clean outs.

Revolving caps sometimes used on sheet metal hoods become jammed or worn out in time. Sheet metal rapidly corrodes in the pressure of chimney gases and rain, especially so when the metal is subject to chlorine and hydrochloric acid in the gases where solid fuel has been treated with calcium chloride to prevent dust.

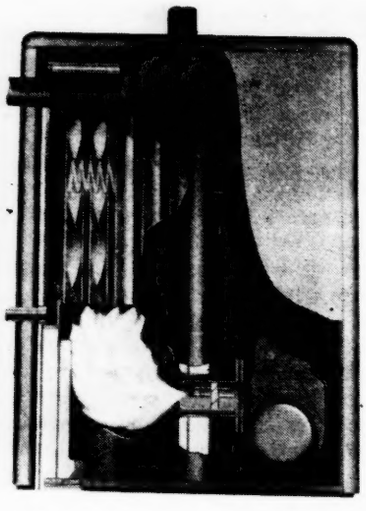
NEWS OF EQUIPMENT AND MATERIALS

Vertical Boiler-Burner

NAME—York-Shipley vertical boiler-burner unit.

PURPOSE—Space heating for small and medium size residences.

FEATURES—Units are offered as a completely redesigned line for use with hot water, steam or vapor type systems. Boiler is oil fired with burner and controls located inside the front casing. A removable panel gives access to the working parts. A hot water coil in the hottest part of the boiler provides domestic hot water without a storage tank. Additional vertical tubes have been added to provide increased heating surface. Entire combustion chamber can be slid out of the front of the unit without disconnecting any piping or raising the boiler. A special coating on the coil exterior protects the coils from electro-chemical action.



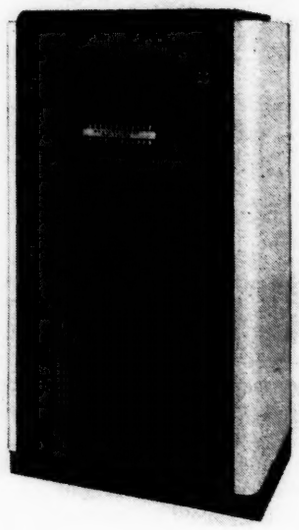
SIZES AND CAPACITIES—Three sizes from 134,000 to 200,000 Btu per hr. Oil firing rate is from 1.35 to 2.0 gallons per hour.

MADE BY — *Residential Division, York-Shipley, Inc., York, Pa.*....406

Unit Air Conditioner

NAME—Westinghouse unit air conditioner.

PURPOSE—Designed for installation in restaurants and retail stores.



FEATURES—Units have been reduced as much as 15% in size as compared to pre-war models of similar capacities and are as much as 20% lighter. Cabinet encloses all the component parts for air conditioning such as the compressor, condenser, evaporator, fan, and glass wool filters. Heating coil and humidifiers may be added for year-round air conditioning. Parts are mounted on a rigid steel chassis with panels easily removable to facilitate servicing.

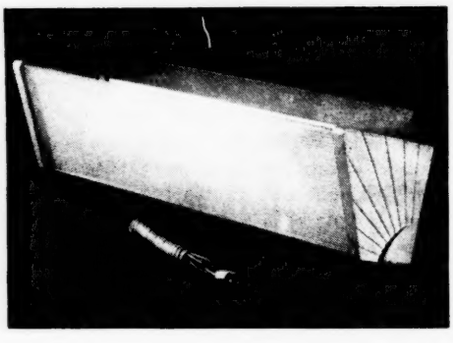
SIZES AND CAPACITIES—The 2-hp model conditions up to 800 cfm; 3-hp up to 1200 cfm, and 5-hp to 2000 cfm.

MADE BY — *Sturtevant Division, Westinghouse Electric Corp., Hyde Park, Mass.* 407

Germicidal Lamp

NAME — Protecto-O-Ray wall unit No. G205.

PURPOSE—Room air sterilization.



FEATURES—Fixture is designed for mounting 6½ to 7 ft above the floor to keep lamp out of the line of vision. Ribbed frosted glass front protects eyes against direct radiation but permits a faint blue glow to show through the glass as a visible indication that the unit is functioning. Unit comes completely wired and equipped with 15-watt Sylvania ultraviolet germicidal tube with the necessary starter, ballast and cord. Lamp is designed for 110 to 125 volts a-c only.

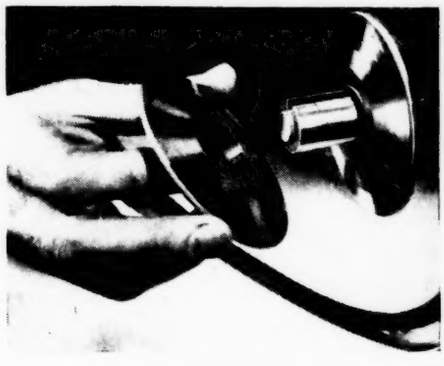
MADE BY — *Lustra Corporation of America, 40 W. 45th St., New York 10, N. Y.* 408

Adjustable V-Belt Pulley

NAME—Spiralok.

PURPOSE—To meet the demand for an adjustable, variable pitch, V-belt pulley.

FEATURES — Pulley locks onto the shaft with which it is used, by means of a spiral wedging action. No set screws are used; no threads or keyways are required on the shaft. Pulley is in rigid alignment when locked on the shaft and an infinite number of adjustments may be easily made. Pulley will drive in either direction of rotation. Makers claim that pulley is particularly adapted for use with air conditioning, refrigeration and ventilating equipment.

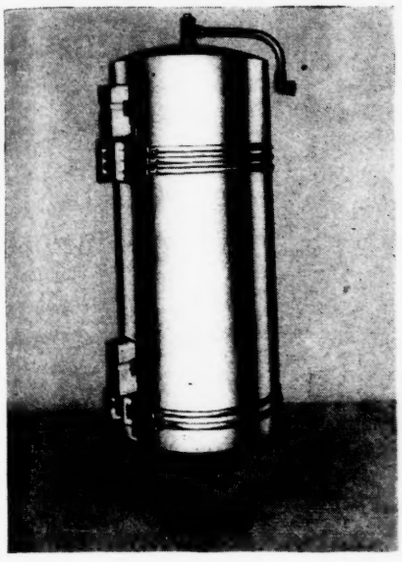


LITERATURE AVAILABLE—Illustrated folder.

MADE BY — *Scientiae Corp., 101 Pine St., Dayton 2, Ohio.* 409

Electric Water Heater

NAME—G-E electric water heater.
PURPOSE—For heating domestic hot water.
FEATURES — Heating element consists of one, two or three ribbons



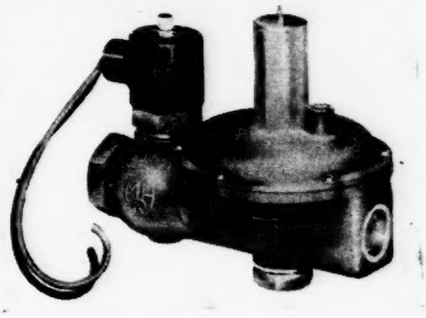
of Calrod that encircle the tank and are held tightly against the surface by stainless steel channels. Maximum heat efficiency is obtained by applying the heat under constant high pressure. To reduce corrosion of galvanized tanks in soft water areas heater can be equipped with a special magnesium alloy tube which sets up a protective electrolytic action. Heaters are insulated with a 3-in. blanket of glass wool. Heat trap in the outlet prevents wasteful circulation of hot water through the house pipes when faucets are not turned on. With the exception of a 30-gal rectangular model all heaters are round.
SIZES AND CAPACITIES—15, 30, 40, 52, 66, and 82 gal with galvanized tanks, and 30, 40, 50 and 80 gal with monel tanks.
MADE BY — *General Electric Co., Bridgeport 2, Conn.* **410**

Foam Destroyer

NAME—DC Antifoam A.
PURPOSE—For destroying foam in aqueous systems.
FEATURES—Substance is a colorless, translucent silicone compound which remains effective in steam distillation or vacuum evaporation. It is lighter than water and is completely insoluble in the aqueous phase of colloidal systems.
LITERATURE AVAILABLE—Illustrated folder which outlines various methods for using this silicone antifoam agent.
MADE BY—*Dow Corning Corp., Midland, Mich.* **411**

Gas Control

NAME—Minneapolis Honeywell gas controller Model V482 and V882.
PURPOSE—Gas valve and control for domestic gas burners.
FEATURES — Instrument was designed to contain a magnetic gas valve with weight and spring-loaded combination regulator. Diaphragm is of type N synthetic buna with a nylon insert. Combination control designed of light-weight materials adaptable for various



types of gases. Unit will be available late in 1947.
SIZES AND CAPACITIES — Available for ½ and ¾-in. pipe sizes.
MADE BY — *Minneapolis-Honeywell Regulator Company, Minneapolis 8, Minnesota.* **412**

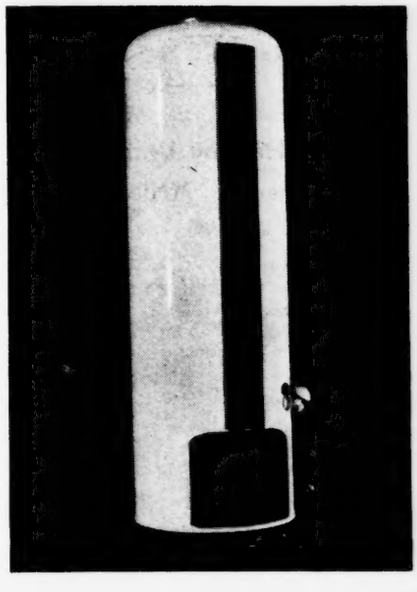
Powered Roof Ventilator

NAME—Ject-O-Valve.
PURPOSE—Automatic operation of roof type ventilator.
FEATURES — Straight-through type of powered ventilator which uses a propeller type fan to exhaust heat, smoke and fumes through a divided top which opens and closes automatically. Ventilator is weather-tight at all times.
LITERATURE AVAILABLE — Bulletin No. 323.
SIZES AND CAPACITIES — Made in five sizes with a wide range of capacities.

MADE BY — *The Swartwout Company, Cleveland, Ohio.* **413**

Water Heater

NAME—Coleman automatic water heater.
PURPOSE—For supplying domestic hot water.
FEATURES—Models have automatic fuel control, automatic pilot and temperature control. The vaporizing type burner is designed for easy removal for cleaning.
LITERATURE AVAILABLE—Catalog 14.
MADE BY—*The Coleman Co., Inc., Wichita 1, Kans.* **414**

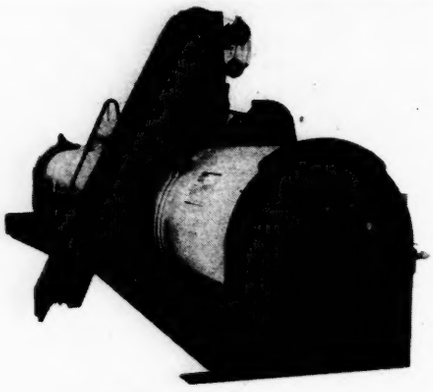


Farm Dehydrator

NAME—The Challenger.

PURPOSE — Application of oil-fired heating and power plant for drying and storing perishable farm feed products.

FEATURES—Drying is accomplished by a long revolving cylindrical

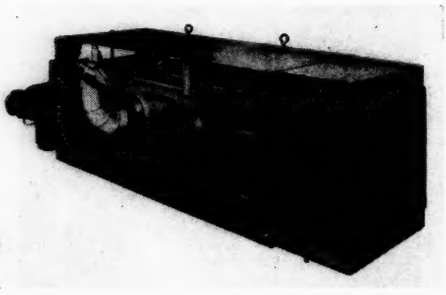


drum through which furnace heat is drawn by an exhaust blower. The raw feed product is fed into a hopper near floor level and is conveyed to the shredder where revolving corrugated knives cut up the material and feed it into the upper end of the dehydrator drum. In this drum the drying products are fed gradually to the separator at the outlet end where they are ejected for cooling, sacking, and storage. Unit is said to be suitable for dehydrating alfalfa products, root crops, and similar types of livestock feed. Dehydrator is fired by a York-Shipley industrial type rotary burner using heavy fuel oil.

MADE BY—*J. B. Beaird Co., Inc., Shreveport, La.* **415**

Heating Unit

NAME—Superfex Model 550 suspended oil furnace.



PURPOSE — To provide a heating unit that can be suspended from ceiling so as to avoid the use of floor space.

FEATURES — Model, which weighs 650 lb, can be suspended from existing ceiling beams or girders without additional structural support. Furnace is entirely automatic in operation and an axial flow blower provides circulation of heated air to the space below. Fuel to operate the burner is drawn from an outside storage tank by means of a two-stage fuel pump on the burner. Unit has a rated output of 200,000 Btu per hr. Filters are installed to remove dust from the heated air.

MADE BY — *Perfection Stove Co., Cleveland, Ohio.* **416**

Welding Electrode

NAME—Airco No. 374 electrode.

PURPOSE — For producing machinable welds on cast iron.

FEATURES — Electrode has a high nickel core wire and a heavy extruded coating that facilitates use on castings requiring machinable welds. Ordinarily preheating is not necessary. Makers claim that multiple pass welds can be made without danger of cracking. Electrode may be used on either a-c or d-c.

SIZES AND CAPACITIES — Made at present in 5/32- and 1/8-in. diameters. Plans are to produce 3/32- and 3/16-in. diameters if demand warrants.

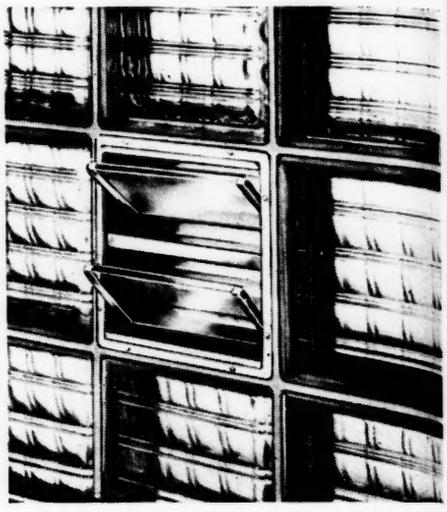
MADE BY—*Air Reduction Sales Company, 60 East 42nd Street, New York 17, N. Y.* **417**

Ventilator

NAME—Weather-Bloc.

PURPOSE—Provision for ventilation through wall of glass block construction.

FEATURES—Made with an aluminum body, the ventilator fits into the space formerly occupied by a glass block. Unit has glass front diffusers and inside glass louvers. Front and rear frames are removable and may be turned or replaced easily. Glass channel weather strips, screws and pivots are of stainless steel. Directional air flow may be upward or downward, or from left to right. Fine mesh screen is placed between the louver and air directors.



LITERATURE AVAILABLE — 12-page catalog.

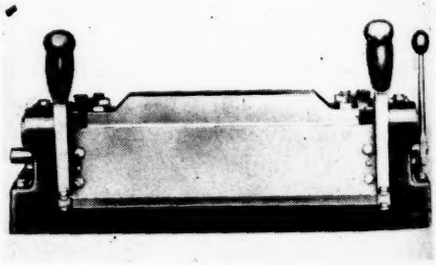
MADE BY—*Air Rectifiers, Inc., 3734 Southport Avenue, Chicago 13, Illinois.* **418**

Universal Brake

NAME—Di-Acro brake.

PURPOSE — Forming machine for sheet metal.

FEATURES—Unit has a double-edge forming blade with both a wide and narrow working surface to allow the formation of very close reverse



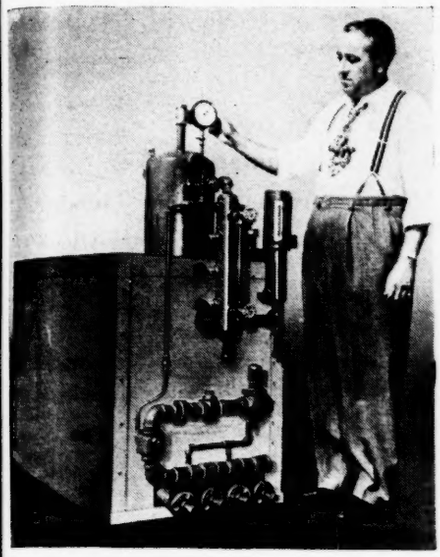
bends. Two different handle positions permit the operator selection of operation for convenience and greater production.

LITERATURE AVAILABLE — Catalog No. 46-11.

MADE BY — *O'Neil-Irwin Mfg. Co., Lake City, Minn.* **419**

Steam Generator

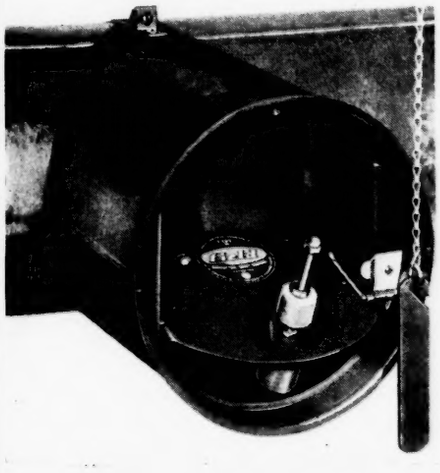
NAME—Model A Steamaster.
PURPOSE — Rapid generation of steam.
FEATURES—Makers claim that steam is produced in less than 10 minutes.



Safety features of the boiler include automatic feed water and steam pressure controls as well as a low water fuel cut off to prevent the firing of a dry boiler.
SIZES AND CAPACITIES—3½, 5, 7½ hp. Larger sizes available in other models.
MADE BY — *Steamaster Automatic Boiler Co., 5819 S. Compton Ave., Los Angeles, Calif.* 420

Draft Control

NAME—Barocheck.
PURPOSE — Automatic barometric draft control and check damper.



FEATURES—Combines the features of the company's draft control, and a check damper in one unit. Makers claim it eliminates dangerously high flue pipe temperatures and wasted fuel. Control can be used with damper motor or with hand operation.
MADE BY—*Field Control Division, H. D. Conkey & Company, Mendota, Ill.* 421

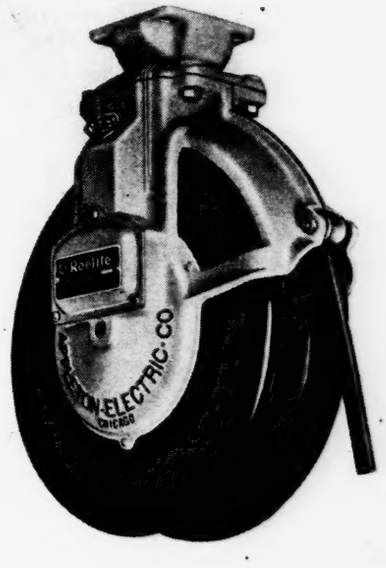
Draft Gauge

NAME—Model MZF draft gauge.
PURPOSE—For draft measurement on every type of heating plant.
FEATURES—Gauge is supplied with 9 ft of rubber tubing with a 5-in. metal sampling tube so that draft measurements may be made at some distance from the point of sampling. Instrument is designed for free movement of pointer at

zero and can be installed on a wall or placed on a flat surface.
LITERATURE AVAILABLE — Leaflet 705.
MADE BY — *Bacharach Industrial Instrument Co., 7000 Bennett St., Pittsburgh 8, Pa.* 422

Cable Reel

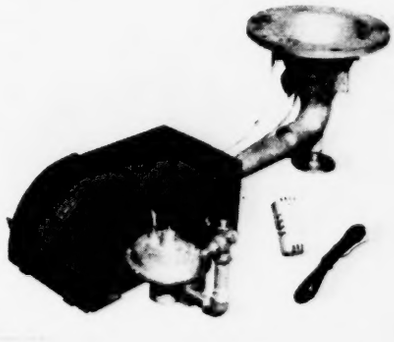
NAME—Appleton take-up reel.
PURPOSE—For paying-out and taking-up electric wire.
FEATURES — Unit is designed to work with a traveling electric hoist or for other uses where electric cable has to be paid out and then taken up on a reel. Conductor cord is kept taut at all times. Oil-less bearings are used at all points of rotation.



MADE BY — *Appleton Electric Co., 1701-59 Wellington Ave., Chicago 13, Ill.* 423

Gas Conversion Burner

NAME — Flint automatic heating unit.
PURPOSE — Gas conversion burner adaptable to any type furnace.
FEATURES — Unit is said to have an efficiency in excess of 80% and is rated from 80,000 to 165,000 Btu. Burner comes supplied with Minneapolis Honeywell controls. Burners can be used with warm air, steam, or hot water heating systems.
MADE BY—*Robert's Manufacturing Co., Flint 1, Mich.* 424

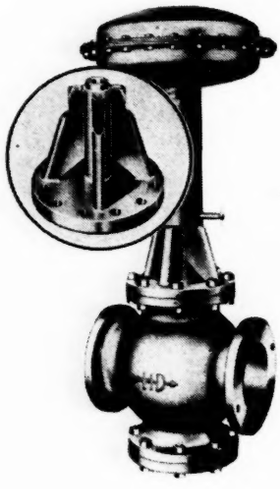


Fin Radiation Bonnet

NAME — Hammel-Dahl vertical U type fin radiation bonnet.

PURPOSE—To provide a natural draft or chimney effect to cool the packing box section of the valves when high temperature fluids are handled.

FEATURES—This type of radiation bonnet is available for the company's line of diaphragm control valves in sizes 2 in. and larger. Makers claim that fins provide 35% better heat dissipation than conventional horizontal fin designs. The fins are fabricated integrally with a forged bonnet and no additional structural support is required. Makers also claim that this type bonnet is equally satisfactory for low temperature applications as its central section is tubular and of exactly the same section that would normally be used in an extension type bonnet. When U fin radiation bonnet is applied to stainless steel or alloy valves, a stainless steel tube and disc of the specified type is retained by a carbon steel housing.



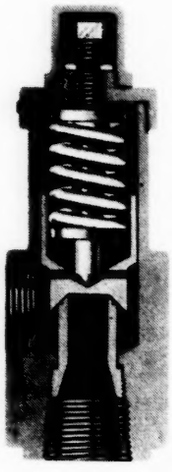
MADE BY—*The Hammel-Dahl Co., 243 Richmond St., Providence 8, R. I.* **425**

Safety Valve

NAME—Farris relief valve.

PURPOSE—Relief valve designed for pressures up to 10,000 lb per sq in.

FEATURES — Valve is fabricated from bar stock, either bronze, cold rolled steel or stainless steel depending on the service for which it is intended. In either case the disc and nozzle are of stainless steel. High carbon steel loading spring is completely enclosed and protected. Nozzle seat is above the drain line and therefore is not subject to accumulations of scale or sediment.



SIZES AND CAPACITIES—Available with or without test gear in ½,-¾- and 1-in. body sizes and interchangeable nozzle inserts.

MADE BY — *Farris Engineering Corp., 493 Commercial Ave., Palisades Park, N. J.* **426**

Water Heater Control

NAME—Type CD108 electric water heater control.

PURPOSE—Hot water control.

FEATURES — Thermostats have a temperature range of 60 to 190F with differentials of 10F for the primary thermostat and 25F for the secondary thermostat. Lower differentials are available on request. Switching is either SPST break on temperature rise or independent circuit double throw. A single screw holds the unit firmly to the bracket that is normally welded to the tank so as to permit adjustment of the conduit box.

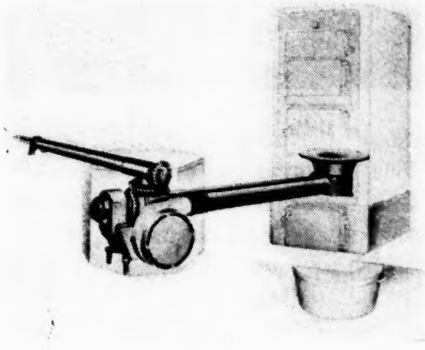
MADE BY—*Cam-Stat, Inc., 2037 So. La Cienega Boulevard, Los Angeles, California.* **427**

Furnace Stoker

NAME—Anchor furnace tender.

PURPOSE — Automatic burning of anthracite coal.

FEATURES—Stoker worm is easily accessible and floats in coal tube without bearings. Unit is designed for slow motion and is operated by precision cut gears and ball bearings. The 1/6-hp motor, rubber-cushioned, has a thermal overload protection. Fire pot operates at low pressure to insure economical combustion. A cast aluminum fan supplies air for the fire bed.



MADE BY—*Anchor Stove & Range Division, Stratton & Terstegge Co., Inc., P.O. Box 1859, Louisville 1, Ky.* **428**

Cast Iron Electrode

NAME—Nicast.

PURPOSE—To provide an electrode to facilitate welding cast iron parts.

FEATURES—Electrodes are made to operate on either a-c or d-c current. Makers claim that it is possible to machine, drill, or tap weld deposits made by these electrodes.

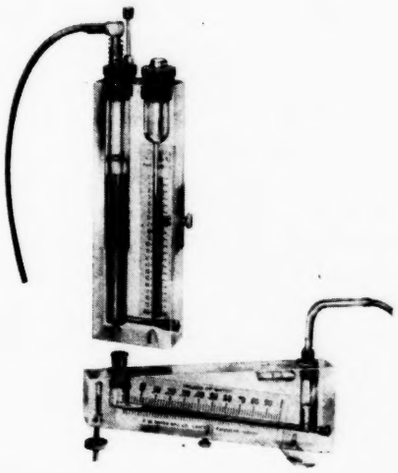
SIZES AND CAPACITIES—Made in ⅛- or 5/32-in. by 14-in. sizes.

MADE BY — *Harnischfeger Corp., Welding Division, Milwaukee 14, Wis.* **429**

Combustion Testing Instruments

NAME—F. W. Dwyer Mfg. Co. combustion testing instruments.

PURPOSE — Transparent CO_2 indicator, and portable draft gauge for increased visibility for line of combustion testing instruments.



FEATURES—Portable draft gauge for measuring all types of air velocities, drafts, and slight pressure changes has light admitted from all sides so that readings are easily observed. CO_2 indicator is made from a solid block of strong, clear plastic with no glass used at all. Metal parts are of stainless steel and instrument is said to have a long life.

LITERATURE AVAILABLE—Catalogue on entire line.

MADE BY—*F. W. Dwyer Mfg. Co., 317 So. Western Ave., Chicago 12, Ill.* **430**

Heat Exchanger

NAME—Karbate impervious graphite shell and tube heat exchanger.

PURPOSE — For use under highly corrosive conditions.

FEATURES — Karbate tubes in bundles are encased in standard 6 in. I.D. steel pipe shells. They will carry temperatures up to 338F and a working pressure of 50 lb per sq in. on both tube and shell sides. Graphite is resistant to most acids, alkalis, and other corosive, solvent or reducing agents. Units can be employed as heaters, coolers, boilers or condensers, and can be operated vertically or horizontally.

SIZES AND CAPACITIES — In three standard sizes of 4 ft 3-in., 7 ft 3-in., and 10 ft 3-in. lengths.

MADE BY — *National Carbon Co., Inc., 30 E. 42nd St., New York, N. Y.* **431**

Solar Heat

NAME—Skylight solar heating unit.

PURPOSE—To provide solar heat for the home or other buildings by means of a special skylight unit.

FEATURES — Skylight unit consisting of an aluminum frame is 30 ft long and 6 ft wide with a flat glass top and sidewall glass sections one foot high. Water pipes that are placed horizontally on the roof under the skylight provide hot domestic water by solar heat. For cloudy and rainy days an electric calrod unit is provided as a secondary means of heating the house. Unit comes crated to be assembled by the heating contractor.

MADE BY—*R. T. Lipscombe, Sr., The Tuckahoe, Cary St. Rd., Richmond 21, Va.* **432**

Air Cooler

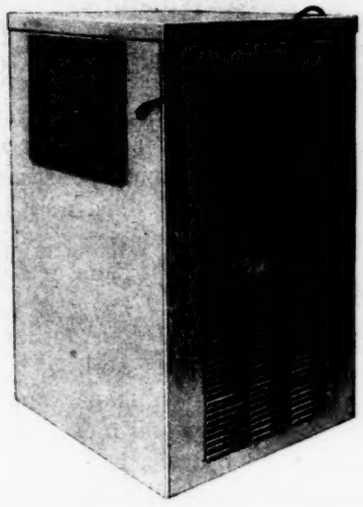
NAME—Polaire air cooler.

PURPOSE — Space cooling in hot, arid sections of this country.

FEATURES—Unit consists of a sheet steel cabinet which houses a blower fan, a revolving pad of aspen excelsior, a 1/3-hp motor to operate both the blower and the pad, and a water reservoir. The pad revolves 4 rpm through the water so that the pad is constantly moist. The blower draws hot, dry air through the roll and the cooled air is discharged into the house.

SIZES AND CAPACITIES—Size to cool a 5- or 6-room house.

MADE BY—*Polaire Cooler Corporation, 141 East Jackson Street, Phoenix, Arizona.* **433**



MAY, 1947

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On the list below, circle the item number of the equipment in which you are interested, using the number found at the end of each item.

406	407	408	409	410	411	412	413	414	415
416	417	418	419	420	421	422	423	424	425
426	427	428	429	430	431	432	433		

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NEWS OF THE MONTH

CONDUCTIVE RUBBER CEILINGS

used in radiant heating of Tennessee test residence. British non-rubber product for similar service now available here.

The home of Harrison J. Hines, of Knoxville, Tennessee, is heated by rubber that conducts electricity. This new heating element is spread over the ceiling of the rooms in thin panels that heat to 110F. The rubber is a wartime development of U. S. Rubber Company and is made conductive by the addition of carbon black. This conductive rubber can be made into decorative screens that heat a room, or in rolls that can be spread in baths or bedrooms lacking sufficient heat.

• **TEST HOUSE.**—The rubber panels used in the test house are four feet square and are installed in the same manner as any other type of ceiling panel. Electricity is conducted to the edge of the panels by copper wire but there is no wire in the heating areas. The house is a bungalow, three rooms, hall, glass enclosed porch, bath, all heated by the rubber ceiling. Designed by the Fonde Construction Co., it is built of hollow cinder blocks filled with insulation. Eight inches of rock wool insulation are above the ceilings. Fuel bill for the season in this area has been estimated at $72.00

• **BRITISH PRODUCT.**—Another product designed for the same purpose is Dulrae, a flexible pulp material similar to heavy wallpaper and containing a heating element of copper and nickel alloy. Developed in England some years ago, it is now available for the first time in this country, Barrington Le B. Musgrave, vice president of Richard Crittal Radiant Heating, Ltd., reported.

Knoxville bungalow heated by conductive rubber.

Dulrae, which does contain wire elements, comes in rolls from which the requisite lengths may be cut in multiples of two feet. These are stapled to plaster board, then applied to the ceiling with the plaster board on the under side. Flexible connecting wires available at convenient position are then connected to the house wiring system.

Mr. Musgrave said that in England it had proved especially adaptable to very old houses which had never had central heating and in which pipe or duct systems could not be installed without making extensive architectural changes.

FIRM PRICES

promised by at least two large manufacturers in attempting to hold inflation line. Builders believe peak is passed.

Two major manufacturers in the heating and ventilating field have adopted the principle of firm prices in an effort to stabilize builders' estimates, while a poll of contractors indicates a return of confidence in the leveling off of construction costs.

• **ACTION.**—The Trane Co., manufacturers of heating, cooling and air conditioning products, announced April 14 that all orders received for early delivery will be accepted at firm prices. All escalator clauses, under which orders were billed at prices in effect at time of shipment, and which have been in effect until now, have been removed. This has been done, company officials state, as a contribution toward promoting throughout the construction industry a return to a price basis that will permit builders to determine in advance accurate prices of construction.

Action by Trane was preceded on April 11 by a similar move by Buffalo Forge Co. and its subsidiary, Buffalo Pumps, Inc., establishing firm prices on all lines of fans and blowers, power plant equipment, air conditioning equipment, unit heaters and pumps.

Edgar F. Wendt, president of Buffalo Forge, said in his statement announcing the policy that his company believed the best interests of the country will be served by a return to normal pricing procedure. While the company has no assurance that costs of labor and materials may not be subject to increase, they hope that firm prices may become sufficiently general to make their position tenable. Said Mr. Wendt, "There is no doubt that we have a period of adjustment ahead. We merely hope to assist in shortening this period."

• **CONTRACTORS POLLED.**—Fifty-three per cent of 268 general contractors in all parts of the country polled by F. W. Dodge Corp., believe that the peak in

construction costs has been reached. In some areas, notably in New England, the South Atlantic states and the East South Central states, opinion was preponderant, at least 2 to 1, that costs had attained their highest level. Only in the Pacific states was opinion general that the peak had not been reached, and here further cost climbs ranging from 11 to 12% during the current calendar year were expected. The minority, or 126 general contractors, believe that further increases can be expected, the average increase anticipated by them being 8.7 to 10.2 per cent above current levels.

• **MOST PRICES.**—At the same time the poll indicated that prices for most building materials, with the exception of lumber, Portland cement, structural steel and cast-iron soil pipe and fittings, have tended to remain stable since January 1. More than half of the contractors reported prices stable for all except two items: lumber and Portland cement. Lumber costs have declined in some areas with the over-all trend stable to downward, while Portland cement, structural steel, cast-iron soil pipe and fittings have tended to increase since the first of the year, the Dodge survey shows.

Other items, mentioned most often as having remained stable since January 1, in the order of stability reported, are: structural clay tile, brick, building blocks, clay sewer pipe, building plaster, warm-air furnaces, hardwood flooring, and cast-iron radiation.

• **HIGH COST FACTORS.** — General contractors, by extremely high majorities, reported that the following unusual factors contribute to the high costs of building at present:

(1) Labor inefficiency caused by irregular flow of materials, 239 to 22.

(2) Delays in construction owing to inadequate supply of skilled workers, 209 to 39.

(3) Excessive expenditure of time in shopping for and obtaining materials, 197 to 53.

Many general contractors reported that the greatest contributing factors to high present-day costs were the low productivity of building mechanics, the low level of skill of some craftsmen, and the failure of workers to do "a day's work for a day's pay." Respondents in general said that they expected improvement in these retarding factors, many in the next year or two, but the consensus was that improvement would come only when the supply of materials and the numerical and qualitative strength of the labor force was adequate to meet the demands for construction.

• **CONSTRUCTION DROP.**—The consensus, 175 to 66, was that there would be a drop in the current high rate of contract letting before the industry attains continuous recovery. The majority felt that costs would stabilize below present levels; the average expected construction cost drop was 14%, as computed from responses of those anticipating that stabilization would occur at levels below those of the present time.

• **LUMP-SUM BIDDING.**—In the aggregate, the general contractors reported that 34.6 per cent of their present work, by dollar volume, is being done on a lump-sum basis, while the remainder is being done on cost-plus-fixed-fee contracts.

COOLED CRUISER

launched as Navy plans to increase crews' fighting efficiency by air-conditioning ships of the Fleet.

Launching of the cruiser *Salem* at Quincy, Mass., late in March marked the opening of a new era in comfort air conditioning of fighting ships. Air conditioning systems are being installed in two new Navy cruisers under an experimental program designed to increase the fighting efficiency of Naval personnel and to improve living conditions aboard ships of the Fleet.

The Navy has made increasing use of shipboard air-cooling equipment for more than 40 years, culminating in the experimental installations now being made aboard the 17,000-ton heavy cruisers, *Salem* and *Newport News*. Except for the machinery spaces, virtually all living and working compartments aboard the new cruisers will be air-conditioned.

• **EFFICIENCY.** — Citing wartime experience, Vice Admiral E. L. Cochrane, U.S.N., Chief of the Bureau of Ships, said "the fighting efficiency of our Fleet is lowered in proportion to the human strain and exhaustion caused by exposing personnel to long periods of abnormal heat. There can be no question but that the crew in an air-conditioned warship will have a distinct and perhaps a decisive advantage in action with an enemy fatigued and of lowered alertness from living in a ship that is not air-conditioned."

Under the experimental program, two types of air-conditioning equipment are being tested by the Bureau of Ships. One, utilizing steam jet equipment, is aboard the *USS Salem* at the Bethlehem Steel Co. Shipyard, Quincy, Mass. The *USS Newport News*, now under construction at the Newport News Shipbuilding and Drydock Co., at Newport News, Va., and scheduled for completion in late 1947, will have equipment utilizing high-speed Freon compressors. Both cooling systems have a simplified Navy coil design which is shockproof and easily cleaned.

"This installation is experimental only insofar as the equipment itself is concerned," Vice Admiral Cochrane noted. "The rigors of tropical warfare pose a tactical problem that can be solved only by air-conditioning all of our major combatant ships in the future. For the present we are seeking to determine what type of equipment is most efficient for the purpose."

• **TESTS.**—Early reports from the South Pacific fighting fronts made it clear that heat exhaustion was a tactical factor in ship design, Vice Admiral Cochrane added. Later in the war, the Bureau of Medicine and Surgery, in cooperation with the Bureau of Ships, investigated the need for shipboard air conditioning and determined the technical requirements for optimum results. Its research clearly established that marked improvement in the efficiency and health of ship's crews would result from use of cooling equipment in living quarters, which would permit more restful sleeping and prevent certain skin ailments caused by heat and humidity.

Extensive experiments were conducted at the Naval

Medical Research Institute, Bethesda, Maryland, and aboard the *USS Franklin*, aircraft carrier, on her shakedown cruise in the Caribbean.

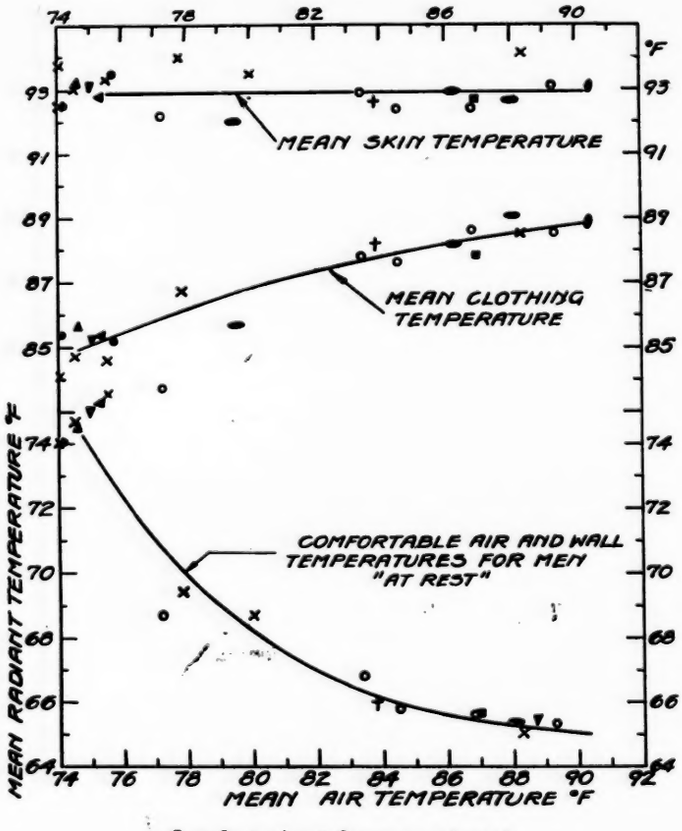
It was concluded that mechanical ventilation, using supply and exhaust fans only, was incapable of producing and maintaining the physiologically acceptable level of 78 degrees effective temperature, and that the cooling and drying of air was most desirable in the interests of higher operating efficiency for personnel aboard ships in tropical climates.

RADIANT COOLING

investigated in tests at Harvard and discussed at New York meeting by Dr. Yaglou.

An experimental study of physical, physiological and comfort aspects of "radiant" cooling was described by Professor C. B. Yaglou of the Harvard School of Public Health at the April meeting of the New York Chapter of the American Society of Heating and Ventilating Engineers.

• **OBJECTIVE.**—The objective of the tests described by the speaker and conducted by him and Dr. Oscar Schneider some years ago, was to determine whether comfort could economically be maintained with high summer temperatures and humidities and without entrance and exit "shock". Tests were carried out in a small wall-cooled room, built inside a larger room which was conditioned by conventional methods. Chilled water, in some cases near freezing, was circulated through coils embedded in three of the walls of the experimental room. The tests were run with dry bulb air temperatures between 75F and 95F and with relative humidities between 40% and 70%. Subjects entered the wall-cooled cubicle from the outer "hot" room which had no cooled walls; they remained in the cool room for two or more hours and then returned to the hot room again. Skin and clothing surface temperatures were recorded at 23 points, and various other measurements were made.


MEAN SKIN TEMPERATURE
MEAN CLOTHING TEMPERATURE
COMFORTABLE AIR AND WALL TEMPERATURES FOR MEN "AT REST"
MEAN RADIANT TEMPERATURE °F
MEAN AIR TEMPERATURE °F

Comfort chart for men at rest.

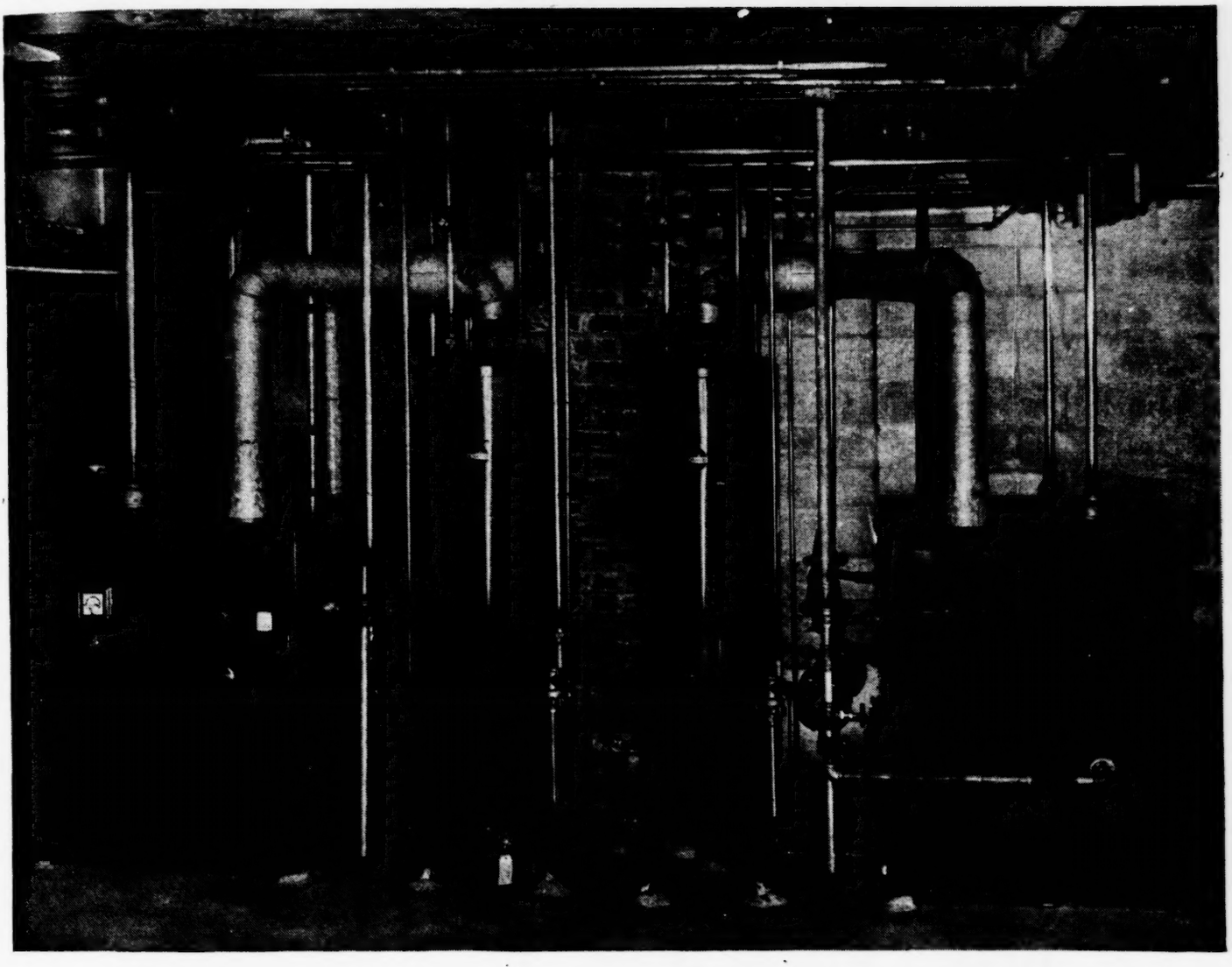
• **COMFORT.**—With the air at 75F, the mean wall temperature required to produce comfort in men wearing customary warm weather clothing, was about 75F, Professor Yaglou stated. As the air temperature was increased, the required mean radiant temperature (wall temperature) had to be lowered until at about 90F the mean radiant temperature was near 65F, as shown in the accompanying figure. The mean temperature of the cooled walls was then 40F and the temperature of the water circulating through the coils was about 35F. With cubicle temperatures of about 89F for men and 92F for women, air temperature had no effect on comfort because the mean "clothing" surface temperature approached the air temperature.

For women, as compared with men, consistently higher (2 or 3 degrees) wall and air temperatures produced comfort, partly owing to their lower metabolic rate and partly because of the considerably lower weight of clothes worn by the women, the speaker reported. For both men and women, regardless of air and wall temperatures, subjects were comfortable when, according to tests, their skin temperature averaged 93F.

• **SHOCK.**—The greatest asset of radiant cooling was a distinct alleviation of thermal shock on entering and leaving the cubicle. The sudden drop of mean skin temperature on entering the cubicle from the outer room kept at 95F — 40% relative humidity, was only 1.5F, when the cubicle walls were cooled, as against 3.5F when the air was cooled to 80F — 50% relative humidity (walls also at 80F). After a two hour exposure, the mean skin temperature approached 93F in both cases.

Under the experimental conditions, the refrigeration load for radiant cooling was from 1.4 to 4.2 times greater than that needed when the cubicle was air cooled to 80F and 50% relative humidity, (walls and air at 80F) by the circulation of conditioned air. The higher the cubicle temperature and humidity, the higher was the load ratio. This unusually high refrigeration load with "radiant" cooling was due to the large quantities of warm and humid air that had to be circulated through the cubicle in order to maintain a substantial wall-air differential for the purpose of the study. As a result there was much unnecessary cooling of air and condensation of moisture on cold walls, whose temperature had to be below the dew point temperature of the air to produce comfort with high summer temperatures and humidities. Without circulation of air, the humidity soon rose to the saturation point, and there was a steep vertical temperature gradient, as high as 30F from floor to ceiling when the outer room was at 95F and the cooled walls

TROUBLE ALWAYS COSTS MORE THAN REVERE COPPER WATER TUBE



There's Less "Bull Work" with REVERE COPPER WATER TUBE

REVERE Copper Water Tube not only gives longer and better service; it also offers many advantages to the skilled journeyman at the time of installation. Hard and soft Type K and soft Type L tube in sizes up to 1" inclusive may be neatly bent to meet installation needs, thus saving many fittings. Where joints are necessary, they are quickly made, even in cramped spaces, with solder or compression fittings. Pipe covering can usually be omitted from relatively short runs without any significant heat losses. And, in addition, you can frequently install Revere Copper Water Tube one size smaller than would be required for rustable pipe.

Add to these installation advantages the long life of Revere Copper Water Tube and its freedom from maintenance costs and you will quickly see why so many heating contractors are now using this tube and why it is so widely used for plumbing lines.

Revere Copper Water Tube is stamped at regular intervals with the Revere name and the type. These marks are more than identification — they are your assurance of full wall thickness and the close gauge tolerances so essential for tight sweated joints.

You can specify or install, in addition to Revere Copper Water Tube, such other long-lived Revere materials as Red-Brass Pipe; Sheet Copper and Sheet Herculoy for tanks, ducts, pans and trays; Dryseal Copper Refrigeration Tube (dehydrated and sealed); Copper oil burner, heat control and capillary tubes.

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at 39F. Introduction of dehumidified air to reduce wall condensation was not attempted.

• **LIMITATIONS.**—In the light of these experiments, it would seem that "radiant" cooling with great air-wall differentials is impractical, particularly when the humidity is high. Wall cooling is also impractical in rooms occupied by many persons, owing to interference of body heat loss by radiation to the cold walls.

WATER IN REFRIGERANT

is detected with an accuracy of one part per million by new infra-red test.

A new technique for measuring the amount of water in Freon fluorine refrigerants and certain other organic gases and liquids with an accuracy of one part in a million was described to the American Chemical Society in a paper by A. F. Benning, A. A. Ebert and C. F. Irwin of the Organic Chemicals Department of E. I. du Pont de Nemours & Co. They told of research which was carried out at the request of Kinetic Chemicals, Inc., a Du Pont affiliate, to further improve the purity of the Freon fluorinated hydrocarbons, widely used as the cooling medium in mechanical refrigerators.

• **WATER HAZARD.**—"In the case of refrigerants, the water content is of special importance," Mr. Ebert said in presenting the paper. "This is because moisture in a refrigerating system may result in freezing at the expansion valves, in the corrosion of metal parts, and in the formation of objectionable sludge, and so lead to faulty mechanical operation. To keep these undesirable effects at a minimum, it is desirable to keep the water concentration below 10 parts per million. This is to enable manufacturers to make refrigerators that can go ten years without servicing."

• **OLD TEST.**—Previously, Mr. Ebert said, the moisture content of these products has been determined by passing a sample through a series of carefully weighed tubes of phosphorus pentoxide, which absorbs the water but not the Freon and then re-weighing the absorption tubes to see how much water had been picked up. At low concentrations of water, this method, in the hands of the average analyst, is accurate only to about two parts per million, and in addition requires four hours for the test.

• **NEW.**—The infra-red technique is quite simple in theory, Mr. Ebert related. Invisible light, with a wavelength of 2.67 microns, or about one ten-thousandth of an inch, is passed through a transparent cell filled with the sample to be tested. The Freon absorbs very little of the radiation at this particular wavelength, but water strongly absorbs this type of invisible light. Thus, by means of a device which measures and automatically records the intensity of the infra-red light before and after it has passed through the test cell, the amount of water present may be read directly from a calibrated scale.

NEW ENGINEERING DESIGN

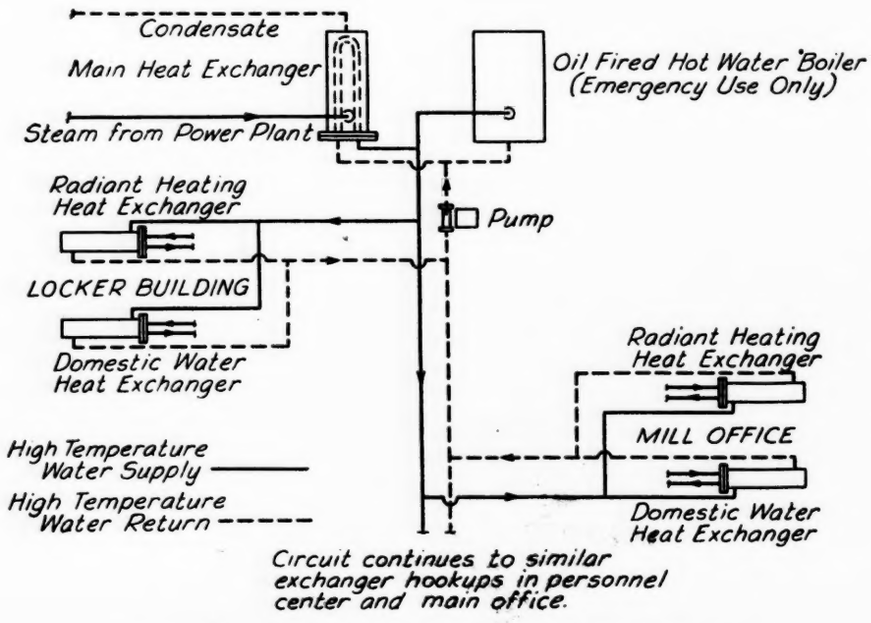
expected to reduce industrial radiant heating costs, uses high temperature water as heat source.

A new design for radiant heating application in industrial plants—aimed to reduce both installation charges and maintenance costs—has been developed by The Rust Engineering Co., Pittsburgh, Pa., for the mill of Container Corporation of America at Fernandina, Fla.

• **HEAT SOURCE.**—The new system, now being installed, utilizes high-temperature water instead of steam as the heat source. It eliminates thermostatic traps, condensate pumps and pressure reducing stations, all of which are factors both in original cost and in the maintenance expense of orthodox, steam radiant heating.

The elimination of these features in the simplified new design, Rust engineers anticipate, will also beneficially affect maintenance by lessening the amount of engineer supervision required. In addition, corrosion, which is likely to occur in steam condensate return lines, is reduced to a minimum, giving another maintenance advantage.

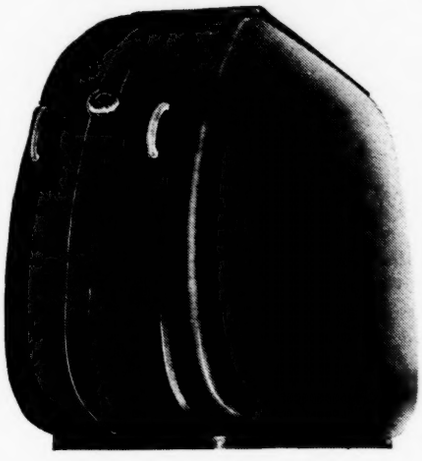
This installation, which is believed to represent the first industrial use of high-temperature water to transfer heat to radiant coils, will serve a group of four buildings. These are the main office, mill office building, personnel center and medical facility, shower and locker building, where radiant heating coils are installed in the floors.


Condensate
Main Heat Exchanger
Oil Fired Hot Water Boiler (Emergency Use Only)
Steam from Power Plant
Radiant Heating Heat Exchanger
Pump
LOCKER BUILDING
Domestic Water Heat Exchanger
Radiant Heating Heat Exchanger
MILL OFFICE
High Temperature Water Supply
High Temperature Water Return
Domestic Water Heat Exchanger
Circuit continues to similar exchanger hookups in personnel center and main office.

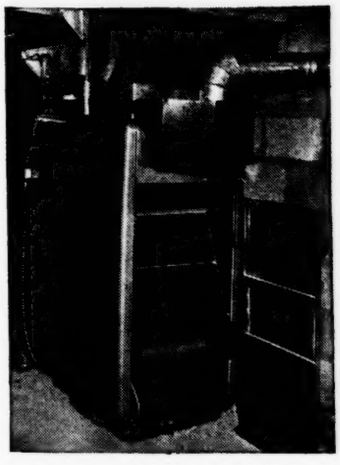
Radiant heating in plant of Container Corp. of America.

Home of Mr. and Mrs. Meyer Feldman, Rochester, New York.
Architect-builder—P. H. Kelly and Sons, Rochester, New York.
Heating Contractor—ABC Sheet Metal Heating and Roofing Co., Rochester, New York.

NO WONDER MORE PEOPLE INSIST ON *Warm-Air Heating*



Mor-Sun Model T-6-G gas-fired furnace, made by Morrison Steel Products, Inc., Buffalo, N. Y. Equipped with 3 DUST-STOP Air Filters.



• Homeowners and potential owners, by the thousands, have expressed their preference for warm-air heating in several recent surveys. They know the "extra" values it gives in comfort, convenience and economy.

More people are insisting on today's winter-air-conditioning systems because of this exclusive combination of values:

1. **WARM AIR,** with room temperatures quickly responding to automatic controls.
2. **CLEAN AIR.** Filtered at the heating unit, all heat delivered throughout the warm-air duct system is free of nuisance dusts, lint and most air-borne bacteria. Housekeeping burdens are lighter because walls and furnishings stay clean longer.
3. **MECHANICALLY-CIRCULATED AIR** keeps warm air fresh and clean while providing the proper number of air changes per hour.
4. **HUMIDIFIED AIR** affords greater physical comfort at lower room temperatures.

Architects, builders and contractors who specify and provide modern warm-air heat know that circulating air will be cleaned efficiently. For DUST-STOP* Filters, a Fiberglas** product, are the choice of most manufacturers as original equipment. They're the homeowner's choice, too, for replacement DUST-STOPS are readily available at low cost through suppliers in every community.

For complete information see Sweet's Files or write: Owens-Corning Fiberglas Corporation, Dept. 912, Toledo 1, Ohio. Branches in principal cities.

In Canada: Fiberglas Canada Ltd., Toronto 1, Ontario.

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TM REG. U.S. PAT. OFF.

MEMBER NATIONAL WARM AIR HEATING AND AIR CONDITIONING ASSOCIATION INDOOR COMFORT

*T. M. Reg. U. S. Pat. Off.
**Fiberglas is the trade name (Reg. U. S. Pat. Off.) of a variety of products made of or with glass fibers by Owens-Corning Fiberglas Corporation.

• **DESIGN DATA.**—Temperature in the grids range from 110 to 130F. Ventilating air is tempered between 70 and 85F by circulating 240F water through hot water Aerofin coils in the ventilating system.

Individual boiler circulating pumps have been provided for each radiant heated area. A series of by-passes permits any pump to be taken out of service temporarily without materially affecting the entire system. This same pump arrangement permits summer operation of domestic heaters using only one pump for the entire system.

Minneapolis-Honeywell Modutrol systems will be used in the four areas. Each system will have an outdoor thermostat connected through an automatic reset panel to control the temperature of water leaving the radiant heat exchanger. Circulators will be controlled by a room thermostat, with a thermostat embedded in the floor to limit slab or surface temperatures to about 90F. Circulators will be cut out when the 3-way mixing valve to the grid supply is in a full by-pass position.

• **OPERATION.**—The operation is easily visualized by means of the accompanying simplified flow diagram. The high-temperature water is brought to the desired temperature by plant steam in a central heat exchanger. An oil-fired hot water boiler is installed for standby operation during a shutdown of the main plant. The heated water circulates to the individual buildings where it is transferred by heat exchangers to water in the radiant coils of the heating system.

The water on the main circuit then returns to the central heat exchanger for heating, and the circuit is repeated. The same high-temperature water will also be used, in this installation, to heat domestic water and to temper air for ventilating systems for the mill.

COPPER CRISIS

threatens as U. S. supply lags far behind production.

A serious situation has developed around the acute shortage of copper in this country, manufacturers using copper state. Production schedules, especially in the fields of electrical and heat transfer manufacture, will have to be reduced with resulting unemployment if supplies are not replenished.

Some companies have appealed to senators and representatives for prompt congressional action for import assistance and the elimination of the excise tax on copper imports to increase domestic stocks.

• **APPEAL.**— F. M. Young, President of the Young Radiator Company, Racine, Wisconsin, has written to Secretary of the Interior J. A. Krug, Assistant to the President Dr. John R. Steelman, Senator Joseph R. McCarthy, and Representatives Lawrence H. Smith, John C. Brophy, Alexander Wiley, John W. Byrnes and Frank B. Keefe, declaring, "The Patterson Bill H. R. 2404 to Relieve the Copper Shortage was passed by the House suspending import tax until April 1, 1949. The bill is now before the Senate Committee on Finance awaiting consideration. A suspension, it is believed, for a minimum of two years of the $80.00 per ton import tax is necessary because it is stated the world market has already largely committed all foreign production throughout the balance of 1947. Foreign producers, therefore, will not make commitments for American import unless they can foresee a reasonably good market ahead."

The Patterson Bill has since passed the Senate and is awaiting presidential signature.

Beginning in early Spring, Mr. Young said, the shortage of copper will begin to have its most serious impact. Major industries throughout the United States face production cutbacks and curtailed operations. Panic bidding for the short supply of copper will force new price rises and will add to the inflationary pressures upon the national economy and diminish the supply of finished products using copper.

• **SUPPLY.**—Domestic consumption of refined copper is running about 130,000 tons a month against production of 80,000 tons—a lag of 50,000 tons. This shortage has been made up during the last few months by drawing from the government stock pile that was built up during the war but is now practically depleted, one company spokesman said.

Imports from Mexico and Chile have also helped to some extent. Until recently, the Government has been buying copper imports and selling this copper to domestic users at the prevailing domestic prices, the Government absorbing the $80.00 per ton excise tax.

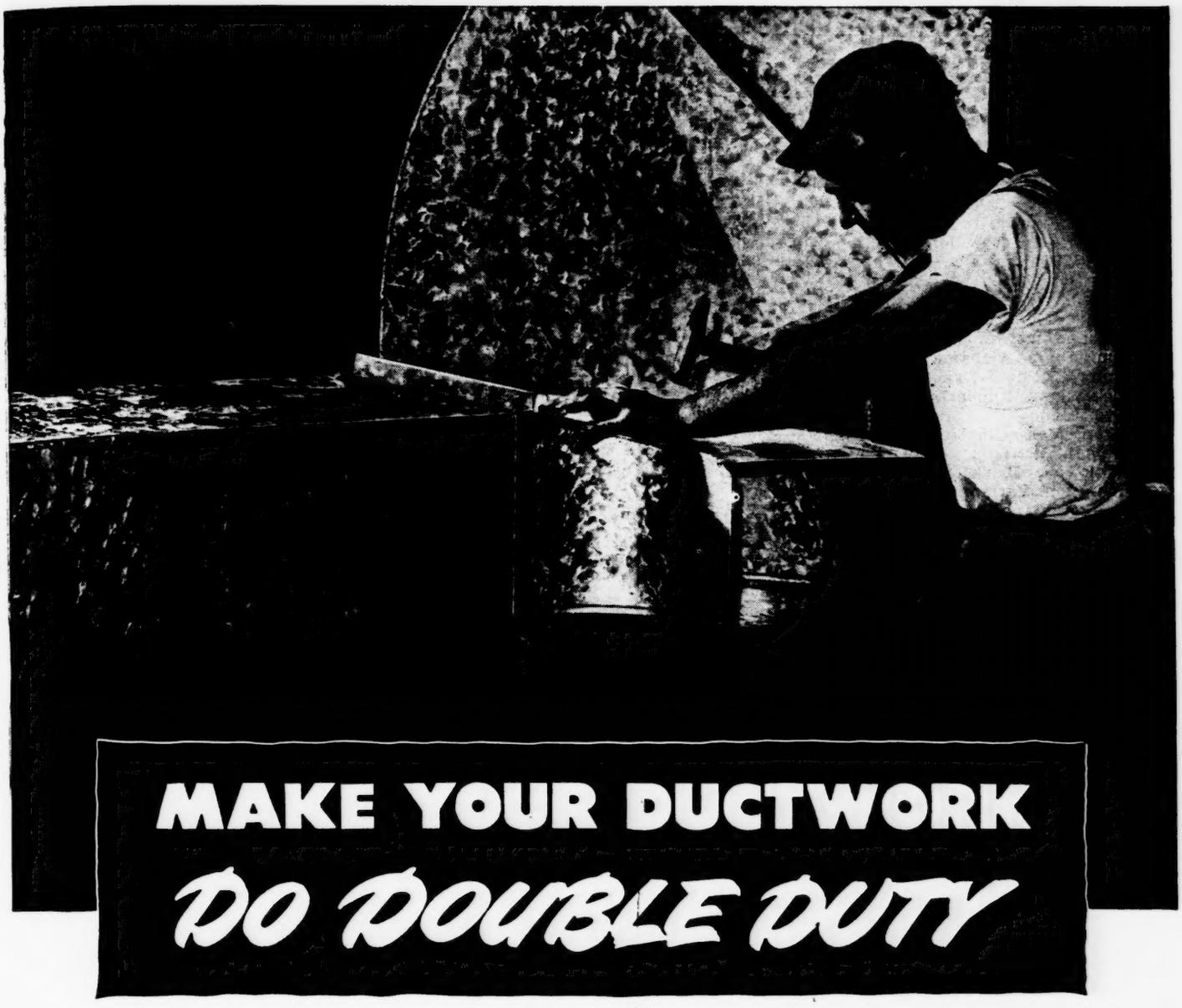
CORROSION TESTS

show how grounded magnesium electrode provides cathodic protection for water tanks.

The life of the usual galvanized-steel electric water-heater tank is greatly influenced by the character of the local water. Some 20 years' service can be expected under ideal conditions from a properly galvanized steel tank. In a few areas—curiously enough these are the areas in which the water is soft rather than hard—galvanized tanks corrode in a small fraction of 20 years. The difference seems to lie in the lack of lime or carbonates in the water, which partially protects the tank wall. Corrosion is largely the result of electrolysis.

• **ELECTRODE.**—Inasmuch as corrosion cannot be eliminated, at least the attack can be diverted to something less vital to the system than the zinc surface. Favorable results have been obtained experimentally by Westinghouse through the simple expedient of suspending inside the tank a magnesium-alloy electrode electrically grounded to the tank. Galvanic currents are such that ions flow away from the magnesium electrode instead of the tank wall, thus the magnesium is destroyed instead of the zinc. When the magnesium electrode is well disintegrated it can be replaced readily.

This manner of preserving the interior follows the well-known principle of cathodic protection, long applied to pipe lines, water tanks, and other buried structures that are subject to galvanic earth currents.



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By double duty we mean ductwork that gives your customer the years of trouble-free service he expects and in turn brings *you* the kind of good-will that builds business.

Ductwork of this quality demands sheets with greater-than-average resistance to rust. Beth-Cu-Loy Galvanized Sheets have this resistance. Their effective life in corrosive atmosphere is 2 to 2½ times that of ordinary galvanized steel sheets.

This added life is the result of two factors. One is the outer protection of the coating of Prime Western zinc, the other is the inner protection of the copper-bearing steel base. Corrosion that eventually breaks through the coating is resisted by the steel itself.

This dual defense against rust makes Beth-Cu-Loy Galvanized Sheets valuable for use in ductwork as well as in other types of sheet-metal applications where the design makes painting impractical.

BETHLEHEM STEEL COMPANY, BETHLEHEM, PA.

On the Pacific Coast Bethlehem products are sold by Bethlehem Pacific Coast Steel Corporation

Beth-Cu-Loy Galvanized Sheets

BETHLEHEM STEEL

Degree-Days for March, 1947

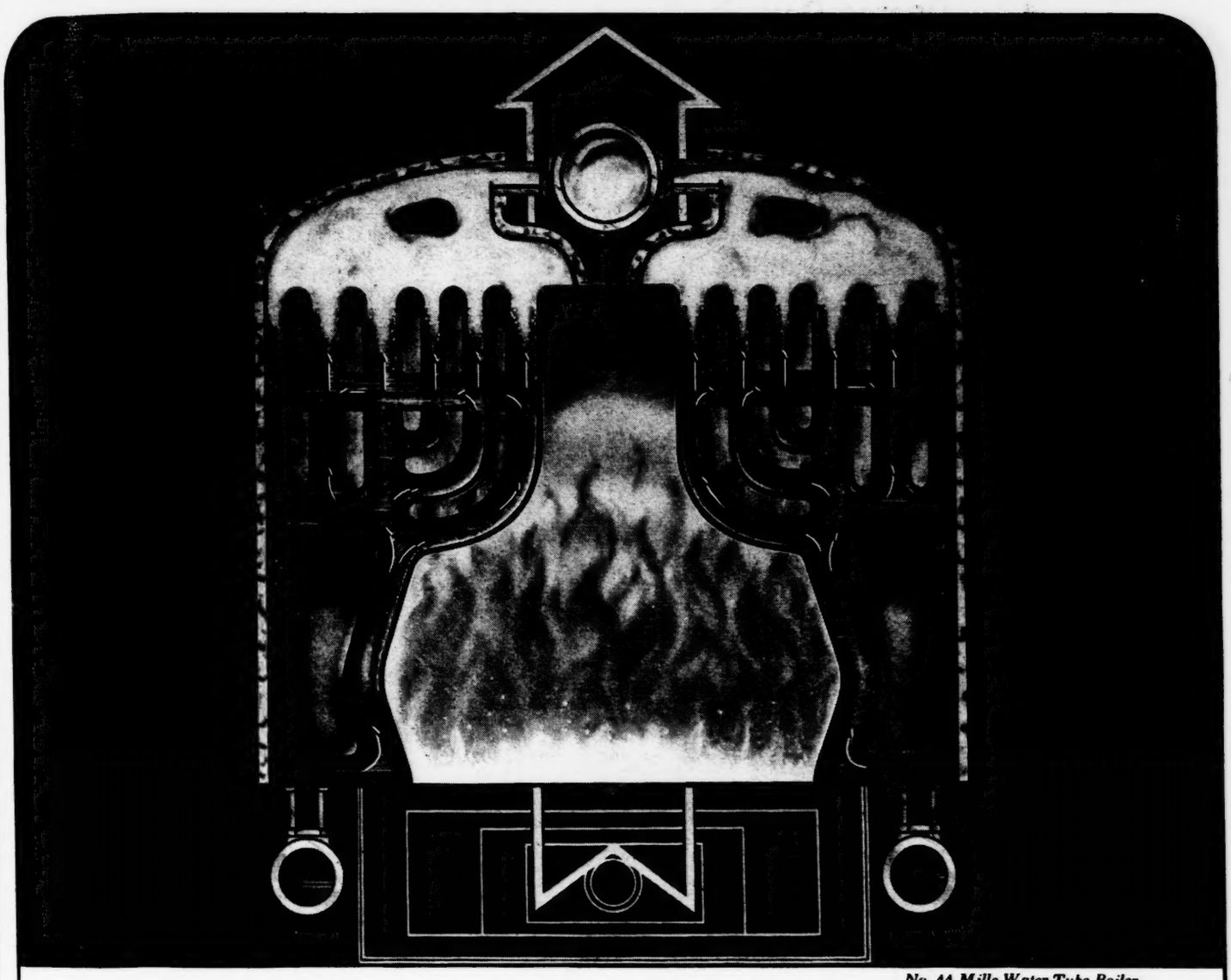
HEATING AND VENTILATING's 20th Year of Publication of Monthly Degree-Day Data

City	March			Cumulative, Sept. 1, 1946 to March 31, 1947			Heating Season[1]
	1947	1946	Normal	1946-47	1945-46	Normal	Normal
Abilene, Texas	469	192	273	2663	2334	2034	2061
Albany, New York	1021	661	980	5698	5858	5848	6580
Albuquerque, New Mexico	543	532	527	3923	4041	3970	4298
Alpena, Michigan	1197	810	1213	6457	6414	6873	8299*
Anaconda, Montana	933	903	1081	6949	6338	6737	8357**
Asheville, North Carolina	806	342	555	3756	3600	3831	4232
Atlanta, Georgia	611	183	388	2754	2654	2758	2890
Atlantic City, New Jersey	848	549	806	3806	3964	4437	5176
Augusta, Georgia	485	128	273	2143	2023	2137	2161
Baker, Oregon	721	797	880	5814	5823	5904	7163
Baltimore, Maryland	764	377	701	3643	3653	4163	4533
Billings, Montana	(a)	711	955	(a)	5610	6209	7119
Binghamton, New York	1052	603	995	5316	5426	5960	6808
Birmingham, Alabama	568	186	260	2658	2552	2283	2352
Bismarck, North Dakota	1221	848	1283	7798	7842	8152	9192
Block Island, Rhode Island	876	687	911	4265	4581	4678	5788
Boise, Idaho	624	659	651	5164	4995	4881	5552
Boston, Massachusetts	847	561	908	4586	4900	5230	6045
Bozeman, Montana	1008	912	1096	6709	6549	6993	8521**
Buffalo, New York	1089	633	1032	5424	5528	5800	6822
Burlington, Vermont	1076	782	1088	6295	6553	6584	7514
Butte, Montana	999	923	1048	7335	6819	6683	8272
Cairo, Illinois	723	196	539	3536	3252	3708	3909
Canton, New York	1127	756	1160	6563	6819	7002	8020
Charles City, Iowa	1078	645	1024	6339	6452	6752	7588
Charleston, South Carolina	438	109	242	1679	1674	1733	1769
Charlotte, North Carolina	640	201	428	2879	2709	2961	3120
Chattanooga, Tennessee	684	230	415	3252	3032	2974	3118
Cheyenne, Wyoming	1056	833	995	6239	5837	6174	7466
Chicago, Illinois	981	562	845	5095	5109	5246	6077
Cincinnati, Ohio	839	299	676	4024	3972	4351	4684
Cleveland, Ohio	1025	503	930	5023	4848	5371	6155
Columbia, Missouri	880	287	694	4397	4114	4597	4922
Columbia, South Carolina	532	140	301	2237	2161	2301	2364
Columbus, Ohio	943	390	778	4610	4461	4891	5398
Concord, New Hampshire	1040	739	1011	6126	6400	6332	7353
Concordia, Kansas	881	393	735	4894	4329	4908	5315
Dallas, Texas	465	146	267	2370	2063	2247	2256
Davenport, Iowa	972	499	893	5312	5345	5718	6289
Dayton, Ohio	1013	433	744	4957	4840	4787	5264
Denver, Colorado	853	575	800	5017	4554	5073	5874
Des Moines, Iowa	998	510	890	5588	5369	5825	6384
Detroit, Michigan	1058	589	973	5319	5298	5691	6490
Devils Lake, North Dakota	1353	1018	1323	8572	8880	8694	9970
Dodge City, Kansas	819	476	688	4651	4139	4646	5035
Dubuque, Iowa	1024	592	958	5768	5814	6152	6790
Duluth, Minnesota	1257	906	1265	7875	7887	7945	9483
Eastport, Maine	994	862	1110	5965	6364	6612	8520**
Elkins, West Virginia	1065	529	763	5082	4794	5019	5697
El Paso, Texas	428	462	282	2252	2233	2383	2428
Ely, Nevada	851	917	(a)	6286	6446	(a)	(a)
Erie, Pennsylvania	1035	572	973	4912	4920	5407	6273
Escanaba, Michigan	1177	867	1262	6833	6808	7290	8771*
Evansville, Indiana	869	275	620	4220	3944	3968	4244
Fort Smith, Arkansas	619	242	397	3159	2937	3054	3147
Fort Wayne, Indiana	1055	524	818	5410	5364	5321	5925
Fort Worth, Texas	478	165	27	2409	2099	2121	2148
Fresno, California	196	314	316	2466	2319	2199	2334
Galveston, Texas	270	65	78	1389	1044	1016	1016
Grand Junction, Colorado	646	599	672	5150	5069	5046	5548
Grand Rapids, Michigan	1052	587	964	5402	5446	5827	6535
Green Bay, Wisconsin	1102	792	1125	6536	6650	6903	7825
Greensboro, North Carolina	765	303	474	3505	3393	3319	3529
Greenville, South Carolina	657	225	446	2936	2866	3185	3380
Harrisburg, Pennsylvania	878	500	803	4517	4569	4880	5375
Hartford, Connecticut	909	599	887	4979	5357	5303	6036
Hatteras, North Carolina	563	231	465	2046	2164	2316	2571
Havre, Montana	1346	742	1169	7725	6627	7557	8700
Helena, Montana	1009	776	1004	6923	6094	6615	7898
Houston, Texas	267	64	65	1456	1185	1157	1157
Huron, South Dakota	1123	760	1092	6915	6931	7167	8004
Indianapolis, Indiana	944	366	756	4592	4493	4855	5298
Jackson, Mississippi	434	159	(a)	2180	2043	(a)	(a)
Kansas City, Missouri	848	300	691	4400	4043	4644	4956
Kewanee, Illinois	1021	485	884	5360	5126	5524	6139
Knoxville, Tennessee	738	228	496	3451	3173	3460	3670
La Crosse, Wisconsin	1197	630	1033	6586	6661	6611	7322
Lander, Wyoming	980	805	1028	6711	6447	6634	7947

[1]Figures in this column are normal totals for a complete heating season, September to June, incl.

Figures in this table, with nine exceptions, based on local weather bureau reports. Exceptions are Utica, Lewiston, Kewanee, Sioux City, and Jackson, figures for which are furnished through the courtesy of Coke Sales Department, Central New York Power Corp., Utica, N. Y.; Norman E. Ross, Bursar, Bates College, Lewiston, Me.; J. M. Hartman, Engineering Department, Kewanee Boiler Corp., Kewanee, Ill.; Alfred R. Wagstaff, Engineer, Sioux City Foundry and Boiler Co., Sioux City, Iowa, and H. M. Ludlow, Box 1368, Jackson, Miss., respectively; Anaconda, Bozeman, Butte and Livingston, Mont., through the courtesy of the Montana Power Company.

[Table concluded on Page 110]



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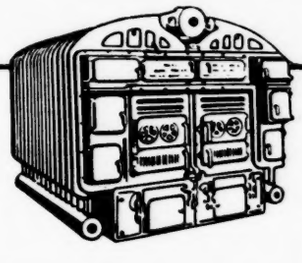

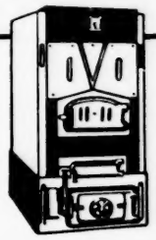

H.B. Smith

CAST IRON BOILERS

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Degree-Days for March, 1947 (Concluded)

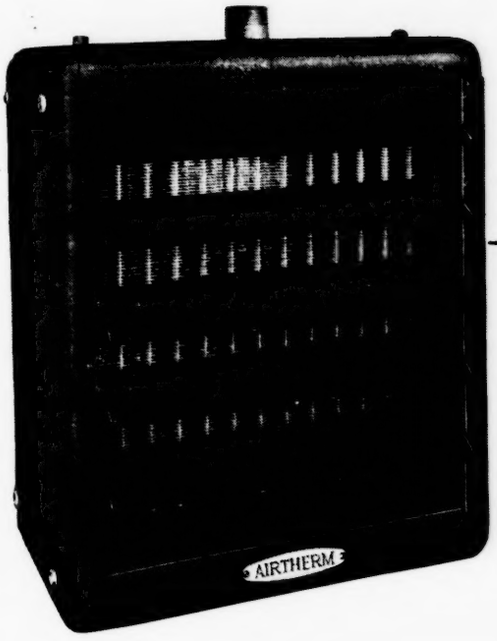
HEATING AND VENTILATING's 20th Year of Publication of Monthly Degree-Day Data

City	March			Cumulative, Sept. 1, 1946 to March 31, 1947			Heating Season[1]
	1947	1946	Normal	1946-47	1945-46	Normal	Normal
Lansing, Michigan	1119	632	1035	5749	5795	6237	7048
Lewiston, Maine	991	800	1093	6208	6643	6625	7707
Lincoln, Nebraska	902	425	840	5184	4751	5483	5999
Little Rock, Arkansas	599	203	353	2960	2746	2733	2811
Livingston, Montana	1037	804	967	6471	5734	5985	7205
Los Angeles, California	169	248	239	1140	1173	1255	1504
Louisville, Kentucky	817	252	589	3898	3615	3916	4180
Lynchburg, Virginia	825	372	564	3806	3693	3707	3980
Macon, Georgia	482	141	260	2225	2094	2168	2201
Madison, Wisconsin	1070	676	1054	6141	6256	6611	7429
Marquette, Michigan	1188	845	1249	6797	6788	7164	8693*
Memphis, Tennessee	619	194	384	3008	2865	2854	2950
Meridian, Mississippi	463	148	245	2267	2063	2136	2160
Milwaukee, Wisconsin	1085	703	1023	6031	5883	6208	7245
Minneapolis, Minnesota	1122	696	1094	6844	6924	7044	7850
Modena, Utah	722	783	827	5384	5436	5591	6562
Montgomery, Alabama	417	110	208	1921	1889	1884	1884
Nantucket, Massachusetts	892	696	899	4469	4585	4811	5957
Nashville, Tennessee	733	198	477	3407	3210	3327	3507
New Haven, Connecticut	914	663	899	4763	5091	5129	5895
New Orleans, Louisiana	249	64	59	1216	1105	1024	1024
New York, New York	851	475	771	4164	4274	4608	5274***
†Nome, Alaska	1430	1811	1660	8999	8757	8886	14580**
Norfolk, Virginia	693	280	521	2793	2785	3104	3350
North Head, Washington	507	625	623	3736	3794	3755	5452**
North Platte, Nebraska	940	620	875	5555	5025	5681	6366
Oakland, California	296	417	363	2522	2530	2392	3143**
Oklahoma City, Oklahoma	596	280	465	3326	2994	3451	3613
Omaha, Nebraska	1213	877	868	4481	4670	5633	6131
Oswego, New York	1049	716	1060	5478	5683	5993	7088
Parkersburg, West Virginia	896	343	670	4203	4012	4371	4775
Peoria, Illinois	1005	460	862	5302	5219	5535	6109
Philadelphia, Pennsylvania	808	421	685	3886	3974	4227	4737***
Phoenix, Arizona	82	119	152	1257	1467	1405	1405
Pittsburgh, Pennsylvania	1008	385	787	4776	4285	4734	5235
Pocatello, Idaho	754	800	843	5791	5979	5696	6655
Portland, Maine	1010	789	1017	5992	6393	6105	7218
Portland, Oregon	402	541	558	3427	3446	3732	4469
Providence, Rhode Island	822	526	890	4480	4730	5206	6015
Pueblo, Colorado	830	618	732	5196	4791	4872	5514
Raleigh, North Carolina	664	258	446	2920	2887	3051	3234
Rapid City, South Dakota	1198	727	1010	6583	5675	6150	7118
Reading, Pennsylvania	848	464	794	4288	4388	4876	5389
Red Bluff, California	269	363	(a)	2427	2549	(a)	(a)
Reno, Nevada	675	752	753	4867	4844	4902	5892
Richmond, Virginia	727	320	543	3303	3317	3449	3695
Rochester, New York	1086	668	1029	5498	5678	5854	6732
Roseburg, Oregon	441	553	546	3624	3384	3636	4428
Roswell, New Mexico	452	343	428	3224	3095	3304	3484
Sacramento, California	277	351	357	2524	2386	2393	2653
St. Joseph, Missouri	886	343	682	4717	4390	4803	5161
St. Louis, Missouri	862	252	657	4099	3846	4309	4585
Salt Lake City, Utah	629	649	722	5129	5070	4866	5555
San Antonio, Texas	297	111	74	1783	1500	1202	1202
San Diego, California	143	243	251	1193	1226	1350	1645
Sandusky, Ohio	998	546	915	4890	4874	5299	6208
San Francisco, California	278	366	353	2177	2069	2123	3264**
Sault Ste. Marie, Michigan	1286	919	1314	7313	7413	7629	9285**
Savannah, Georgia	399	100	167	1554	1540	1490	1490
Scranton, Pennsylvania	991	557	893	5044	5075	5445	6129
Seattle, Washington	462	585	620	3704	3563	3917	4934**
Sheridan, Wyoming	1153	808	1048	6651	5973	6837	8008
Shreveport, Louisiana	428	139	202	2224	2003	1938	1938
Sioux City, Iowa	1044	576	967	6261	5939	6245	6898
Spokane, Washington	703	738	778	5542	5278	5485	6355
Springfield, Illinois	906	325	760	4674	4457	4951	5373
Springfield, Missouri	(a)	328	626	(a)	3891	4149	4428
Syracuse, New York	1058	656	1004	5588	5702	5984	6893
Tacoma, Washington	514	616	648	4001	3881	4086	5181**
Terre Haute, Indiana	944	311	663	4559	4327	4539	4872
Toledo, Ohio	1047	573	905	5323	5298	5372	6077
Topeka, Kansas	833	311	682	4496	4024	4635	4969
Trenton, New Jersey	839	488	735	4184	4325	4450	4933
Utica, New York	1062	1390	989	5129	6610	5955	6796
Valentine, Nebraska	1103	669	980	6305	5641	6205	7039
Walla Walla, Washington	442	565	598	4215	3949	4274	4808
Washington, D. C.	769	357	694	3660	3666	4253	4626
Wichita, Kansas	787	357	648	4261	3821	4421	4673
Williston, North Dakota	1311	844	1280	8033	7858	8238	9323
Winnemucca, Nevada	654	751	767	5177	5177	5411	6427**
Yakima, Washington	551	621	620	5057	4591	5053	5599

(a) Data not available.
†Nome data are for February.

*Includes August.
**Includes July and August.
***New 48-year normal covering 1898 to 1946.

[1]Figures in this column are normal totals for a complete heating season, September to June, incl.



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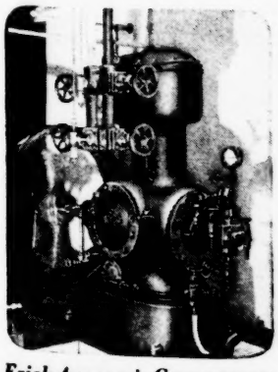

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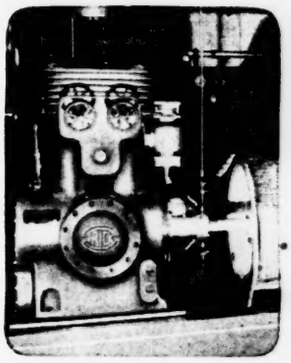

Frick Freon-12 Compressor

Abstracts and Reviews

Publications abstracted in this department should be ordered direct from publisher.

INSULATION OF BUILDINGS

As the result of fuel conservation programs during the war years to save the country's limited supply of fuel, considerable attention has been given to insulation of buildings as a means to cut down on heat loss to outside air. Today, while a timely subject, it is a controversial one because claims made as to building insulation cannot easily be verified.

A collection of important facts and data regarding thermal insulation for dwellings and other structures has been made by Paul D. Close, an engineer who has made a long study of this subject.

After discussing the basic principles of insulation and heat loss coefficients for various types of building construction and materials, he shows how to calculate heat loss, and how to estimate fuel savings as the result of thermal insulation. Beside the author's consideration of dwellings, he also has a chapter on insulating farm buildings other than the farm residence. Farm building insulation is considered important as a means for promoting the health of farm animals and increasing financial returns.

This work should prove valuable to one who wants to develop his knowledge of building insulation.

Included in the book is a tabulation of insulation manufacturers and one on trade names.

Thermal Insulation of Buildings, by Paul D. Close. Cardboard cover, plastic spiral bound, 6 x 9 in., 104 pages. Published by Reinhold Publishing Corp., 330 W. 42nd St., New York City. Price, $1.75.

● ● ●

MOISTURE CONDENSATION—Bulletin F 6.2, part of a series by the University of Illinois on the small home, is an 8-page illustrated booklet of value to the home owner. It covers the evolution of condensation troubles in frame houses, where condensation occurs in the home and corrective measures that should be employed, the problem of basementless houses, the use of vapor barriers, and insulation. A page is devoted to the ventilating of roofs to avoid excessive humidities in the house. A two-page table presents an analysis of places where condensation may occur in a house, the cause of the trouble, and how to element the troubles. For a copy write to Small Homes Council, Mumford House, University of Illinois, Urbana, Ill.

• FINISHES FOR ALUMINUM—Two spiral bound volumes, 6 x 9 in. contains information on the means for applying coating finishes to aluminum. Section 1, of 108 pages covers cleaning treatments, mechanical surface finishes, chemical surface finishes, electrolytic treatments, electroplated coatings, paint and ceramic coatings. Section 2, of 120 pages covers shop data on materials, equipment, solution preparation, procedure and control for the most widely used finishing processes. Price, $2 for the set. Reynolds Metals Co., Dept. 27, 2500 S. 3rd St., Louisville 1, Ky.

HAZARDS IN METAL MINES—As part of Miners Circular 57 on the Health and Miscellaneous Hazards in Metal Mines, there is a 23-page section on the health hazards of metal miners which covers air temperature and humidity, dust, dust concentrations, dust control and sanitation. Copies, 20 cents each, are for sale by the Superintendent of Documents, Washington 25, D. C.

BENZENE—The newest in the series of pamphlets issued by the Division of Labor Standards on controlling chemical hazards, covers the hazards of benzene or benzol. The 24-page booklet contains a chart which points out the fire hazards, storage requirements, fire-fighting phases, and cautions for transporting and handling the fluid. It also contains information on the protection of personnel liable to exposure to benzene or benzene vapors. Free copies may be obtained by writing Division of Labor Standards, U. S. Department of Labor, Washington 25, D. C.

STANDARD FOR TESTING AND RATING CONVECTORS—Commercial standard CS-140-47 issued by the U. S. Department of Commerce for testing and rating convectors, is being circulated as a voluntary standard of the trade to become effective on March 1, 1948. For copies write to the National Bureau of Standards, Washington 25, D. C.

CLEAN AIR — No. 7 in the series on Controlling Chemical Hazards, issued by the U. S. Department of Labor, covers exposure to harmful dusts, fumes, vapors and gases. Written in nontechnical language, the 24-page pamphlet covers the prevention of nuisance and toxic dust exposure in industry and offers suggestions on the types of exhaust systems that may prove of value. For a free copy of this pamphlet write to Wm. L. Connelly, Director, Division of Labor Standards, U. S. Department of Labor, Washington 25, D. C.

COPPER TUBE HANDBOOK—The third edition of the Copper Tube Handbook for Plumbing and Heating, has just been published by the Copper & Brass Research Association, 420 Lexington Ave., New York City. The book is by T. N. Thomson, the author of many books on plumbing, heating and ventilating. Included are chapters for house heating systems, design of a steam heating system, two-pipe hot water system, a one-pipe hot water system and data on radiant heating. For a copy of the 72-page book write to the association.

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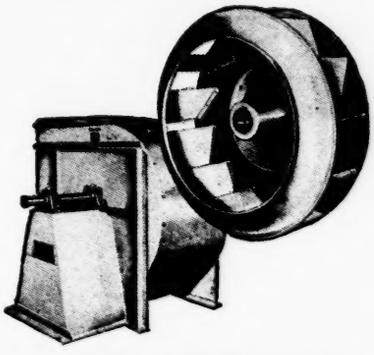

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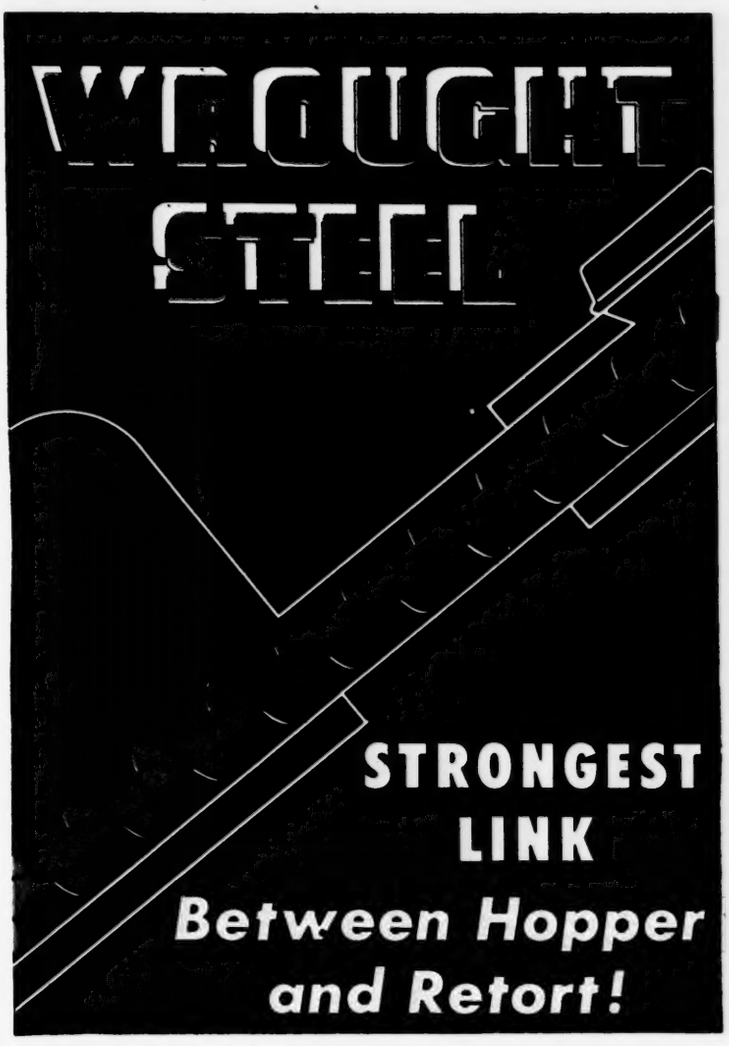

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NEW CATALOGS

Rotary Pumps

Type GA rotary pumps, designed for small capacity and severe service requirements are described and illustrated with selection charts in bulletin W-484-B1. —*Worthington Pump and Machinery Corp., Harrison, N. J.* 252

Stud Welding Data Book

Designed to provide factual information on the use of automatic stud welding, a new 36-page data book and catalog describes and illustrates the Nelson automatic stud welding process, equipment and studs. Information on designing for stud welding is given, together with specifications and physical properties data on Nelson flux-filled studs. A final section of the book is devoted to a description of typical stud welding applications.—*Nelson Sales Corp., Lorain, Ohio.* ... 253

Industrial Cooling Applications

Application of industrial refrigeration and air conditioning is the subject of a 30-page, four-color booklet, illustrated with schematic diagrams of basic heating equipment and photographs of actual installations. The booklet points the way to the most efficient and economical solution to the particular problem accompanying each application. Applications include air conditioning for precision assembly operations, cooling hearth furnace roller bearings, process air conditioning, and so on.—*General Electric Co., Air Conditioning Dept., Bloomfield, N. J.* 254

Steam Equipment Film

Modern steam and mechanical engineering research responsible for improvements in steam auxiliaries is the subject of a 30-minute, color, and sound motion picture, "An Engineering Reason," produced by Hathen Productions, Philadelphia, for Yarnall-Waring Company. Film covers generation and utilization of steam under low, medium and high pressure. Commentary is by Lowell Thomas. Free to engineering and business groups and industrial plants. Request direct on company letterhead.—*Hathen Productions, 264 South Van Pelt St., Philadelphia 3, Pa.* 255

Air Hose

A new 4-page catalog section describes the construction and applications of various sizes and pressure capacities of air hose.—*B. F. Goodrich Co., Akron, Ohio.* 256

Instruments

Bulletin W1811 lists recording thermometers, pyrometers, pressure gages, voltmeters, ammeters and pH recorders.—*The Bristol Company, Waterbury 91, Conn.* 257

Industrial Dust Control

A new 23-page illustrated booklet discusses various types of industrial dust problems and typical applications of AAF air filters to such problems. Included are a chart of duct characteristics and sections on dust control in industrial applications.—*American Air Filter Co., Inc., 215 Central Ave., Louisville 8, Ky.* 258

Pumps

An entirely new catalog of 180 pages covers the Deming line of pumps and water systems. The first 86 pages cover industrial lines of pumps of all types, 65 pages cover the Deming water systems for domestic and farming use, and 29 pages cover pump accessories and spray pumps. Performance and selection tables supplement engineering data relating to the equipment. —*The Deming Company, Salem, O.* 259

Refrigeration Testing

Laboratory facilities for refrigeration testing of commercial and industrial products are described in a 4-page bulletin.—*U. S. Testing Co., Inc., Hoboken, N. J.* 260

Glass, Paint and Metal Products Data

A new edition of the Pittsburgh Data Sheet Handbook is being distributed. This handbook is designed to answer questions concerning the various glass, paint and metal products of the Pittsburgh Plate Glass Co. and Pittsburgh-Corning Corp. Pocket size, looseleaf bound, tab indexed, the folder contains individual sections on each class or product. Each copy is registered in the holder's name, and additional sheets will be issued as new information is available. Copies are available to architects, designers, draftsmen and others in the building trades.—*Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh, Pa.* 261

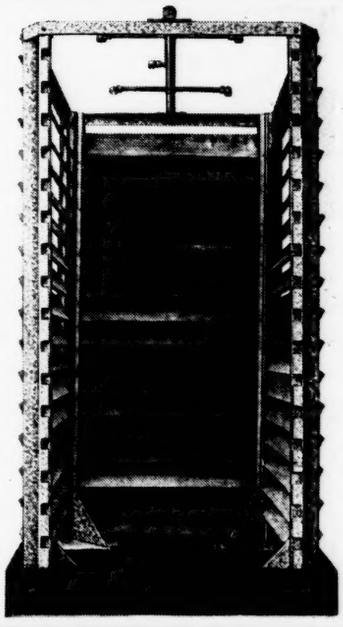
Condensation Meter

A 12-page, illustrated bulletin, No. 35-80B, on the Adsco rotary condensation meter, gives details of construction and installation on seven sizes of meters with capacities from 250 to 12,000 lb of condensate per hour. Meters are used to determine steam consumption by buildings or departments.—*American District Steam Company, North Tonawanda, N. Y.* 262

Thermostats

A four-page folder describing a line of adjustable and non-adjustable bi-metal thermostats has been released. Designated "Type S" thermostats, they have a temperature range from zero to 650F, for 15 ampere, 115 volts a-c operation. The folder suggests applications, describes principle of operation and shows typical calibration curves. In addition to dimension drawings, there are photographs of a number of thermostat styles and construction details.—*Stevens Manufacturing Co., Richland Trust Building, Mansfield, Ohio.* 263

30 STANDARD SIZES OF ATMOSPHERIC SPRAY COOLING TOWERS



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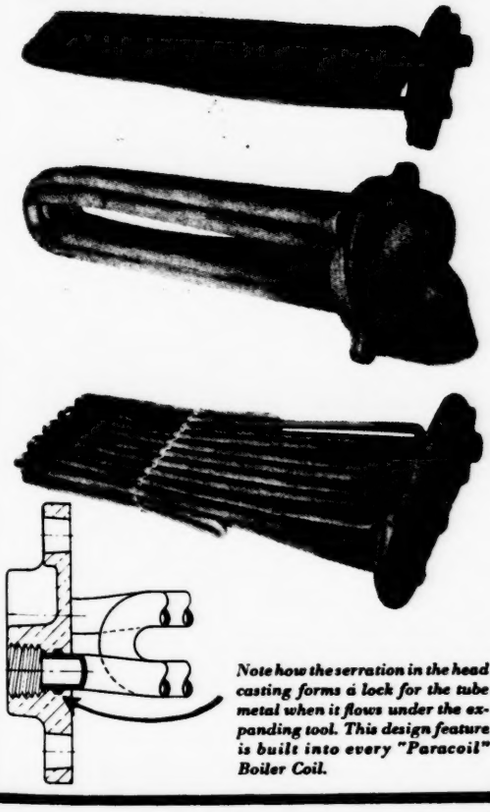

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Merchandising for Dealers

A new booklet on the open-front store, entitled "Open for Business" is offered to plumbing and heating dealers. This publication extends the Crane open front store program inaugurated in 1945. Dealer experience is cited with photographs of actual stores. Special treatment is given to windows and signs.—*Crane Company, 836 South Michigan Ave., Chicago 5, Illinois.* **264**

Air Rectifiers

Air rectifier equipment for air conditioning, heating and ventilating are described in a 12-page catalog. Products include filters and ventilators for industrial and residential applications.—*Air Rectifiers, Inc., 3734 Southport Ave., Chicago 13, Ill.* **265**

The Anthracite Industry

The anthracite industry emphasizes its importance to the national economy with maps, photographs and charts in a booklet describing activities of the Anthracite Institute.—*Anthracite Institute, Wilkes Barre, Pa.* **266**

Solderless Wiring

A tab-indexed, plastic bound, data book on solderless wiring devices displays various types of solderless terminals, their features and applications, with photographs and charts.—*Aircraft-Marine Products, Inc., 1533 North 4th St., Harrisburg, Pa.* **267**

V-Belt Drives

A 144-page, indexed catalog for stock Texrope drives has been issued to simplify V-belt drive selection. More than 22,000 stock drives are listed for all applications from 1 to 150 hp. Conditions specified include horsepower, motor speed, ratio and overload factor.—*Allis-Chalmers Mfg. Co., 1126 S. 70th Street, Milwaukee 1, Wis.* **268**

Hot Process Softeners

A 68-page bulletin describes the Cochrane systems of hot process water treatment for prevention of scale, reduction of silica, and protection against corrosion. Many types of water treatment are described and conditions for the use of each analyzed. Equipment for water treatment is illustrated with drawings and photographs. A section on feedwater chemistry includes chemical formulas and molecular weights of the substances frequently appearing in the chemistry of water conditioning.—*Cochrane Corp., 17th St. & Allegheny Ave., Philadelphia 32, Pa.* **269**

Compressors

Air King compressors, one to 15 horsepower, with pressures to 250 lb per sq in. are the subject of a 4-page bulletin No. H-620-B26.—*Worthington Pump and Machinery Corp., Harrison, N. J.* **270**

Pressure Control Data

Bulletin 462, 20 pages, gives engineering, operating and maintenance data on external-pilot-operated pressure controllers. Design features of pilots and associated valves are illustrated in cross section. Selection charts and capacity tables are provided.—*Leslie Co., 52 Delafield Ave., Lyndhurst, N. J.* **271**

Refrigeration Engineering Data

The Kramer Engineering Data Book, catalog No. R-114, contains engineering data for refrigeration load calculations including wall heat losses and data for special application in food storage with useful conversion and mathematical tables.—*Kramer Trenton Company, Trenton, N. J.* **272**

Stainless Steel Tubing

Bulletin (TB-323) outlines some of the many uses of B & W stainless tubing and briefly describes its important properties from an application point of view. Included are the most commonly used Croloy analyses as well as tubing and pipe size ranges.—*The Babcock & Wilcox Tube Co., 85 Liberty St., New York 6, N. Y.* **273**

Degree Day Influence

"The ABC's of Degree Day Influence on Adequate LP Gas Service" is a booklet compiled to help the liquefied petroleum gas distributors of 20 states in their efforts to promote year round storage of liquefied petroleum gases on consumers' premises. Facts about liquefied petroleum gas and its distribution problem are tied to the degree day formula with an explanation of how storage planning is affected by degree day conditions in a given area. The booklet contains degree day maps and tables.—*Southern Gas & Equipment Co., P. O. Box 378, Tulsa 1, Okla.* **274**

MAY, 1947

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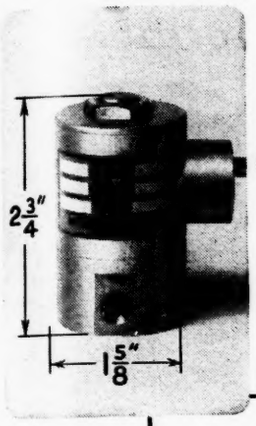

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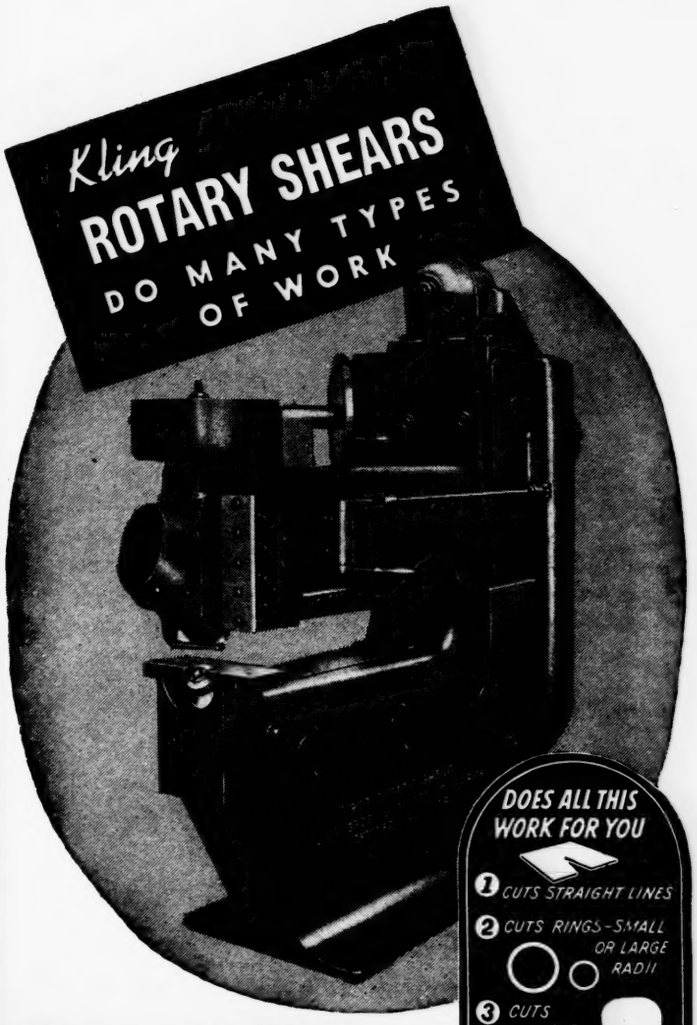
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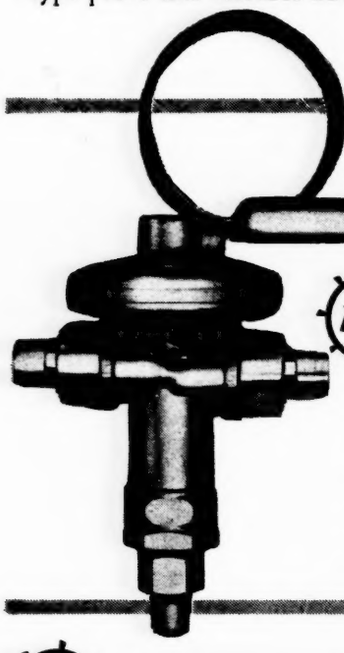

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Getting Personal

Maria Telkes

Maria Telkes (*Solar House Heating—a Problem of Heat Storage*, page 68) received her Ph.D. in physical chemistry from the University of Budapest, Hungary. She was engaged in biophysical research work at the Cleveland Clinic Foundation, Cleveland, Ohio, and in metallurgical research at the Westinghouse Research Laboratories, East Pittsburgh, Pa.

She joined the Massachusetts Institute of Technology's Solar Energy Utilization Project in 1939 as research associate and is engaged in thermoelectric research work in the Department of Metallurgy.

During the war Dr. Telkes studied solar distillation, developing a solar distiller for life-rafts to produce drinking water from sea-water.

W. Parkerson

William Parkerson (*The Coefficient of Performance of the Heat Pump*, page 63), was born in New Orleans, September 23, 1898. He began his college training at Tulane University in 1915, served in World War I from July to December, 1918 and completed his Bachelor of Arts degree at Tulane, in June, 1919. He became an instructor in physics and spent one year each at the University of Pittsburgh and the University of Illinois, receiving his Master of Science degree at the latter university in June, 1922. He returned to Tulane to complete a Bachelor of Engineering degree, which was awarded in June, 1923. After miscellaneous connections, including that of power sales engineer for the Gulf States Utilities Company, he became commercial manager of the Public Service Company of Oklahoma in March 1928. He resigned this position to go into business for himself as the Air Control Engineering Company in New Orleans, representing the Carrier Corporation. In 1940 he joined the Central Power and Light Company in Corpus Christi, Texas, as industrial engineer, transferring to Victoria, Texas, as local manager in 1942. In October, 1945 he joined the Middle West Service Company in Chicago, as rate engineer, the position he now holds. Mr. Parkerson is married, and has three children.

M. J. Druse

M. J. Druse (*Unit Heaters Ideal for Home Basements*, page 89), was born in Milwaukee, Wis. on August 22, 1911. He received his B.A. degree from the University of Wisconsin in journalism and advertising. After early experience in radio network sales and merchandising, he became a partner in the advertising firm of Brady-Druse-Bjorkholm Co. He became associated with Modine Manufacturing Co. in 1935 at which company he is employed as advertising manager. He is a member of A.F.A. and N.I.A.A., is married and has two children. His hobbies are fishing, gardening and photography.

P. L. Davidson

P. L. Davidson *(Refrigeration for Textile Mill Air Conditioning,* page 57), a native of Scranton, Pa., had his college course at Cornell University interrupted by World War I. Following his graduation in 1919, he went with the Buffalo Forge Co., and while there became interested in air conditioning.

He joined the Carrier Engineering Corp. and was appointed assistant manager of the Boston, Mass., office in 1922, followed by promotion to manager. During this period he was closely associated with the air conditioning of textile mills.

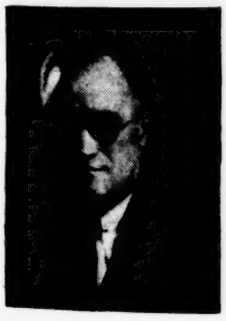
In 1928 he was advanced to manager of Carrier's Philadelphia office. He was responsible for the air conditioning of the 33-story Philadelphia Savings Fund Society Building, the first large commercial building to be completely air conditioned. Since he was in charge of Carrier's operations in the Carolinas and Georgia, he became familiar with the southern textile industry.

Mr. Davidson resigned from the Carrier Corp. in 1936 and opened his own consulting office in Philadelphia, specializing in air conditioning. In 1942 he was retained as air conditioning consultant for a large South Carolina textile mill group and since then has been active in the air conditioning of southern textile mills. In 1945, he opened an office in Greenville, S. C., in charge of John deB. Shepard, formerly air conditioning engineer for Consolidated Gas of Baltimore.

Mr. Davidson is now serving as consultant on air conditioning problems for textile mills from Maine to Georgia.

J. Cassie

Good editorial salesmanship finally overcame the initial request of **John Cassie** to publish his brief biography without his portrait. His biography appears on page 139 of the February, 1947 issue. On page 87 of this issue, there is another of the practical articles *(Minor Changes End School Room Complaints)* by the "fan doctor." Mr. Cassie has a wealth of interesting experiences in treating sick installations. He points out how by simple changes in a system, existing troubles are eliminated or very much corrected. "Doc" Cassie employs unusual means to solve problems, including paper models of an installation made to scale, to study a ventilation or fan problem.



A. F. Davis

A. F. Davis *(Needless Regulations Impede Use of Arc Welding in Industry,* page 85) is truly a native of Ohio. Born in Diamond, Ohio, he attended rural school, Alliance High School, and two years at Mt. Union College. Following a period when he worked for the engineering department of Alliance, he enrolled at Ohio State University in 1911 and in 1914 received the degree of Mechanical Engineer in Electrical Engineering. After graduation, he entered the employ of The Lincoln Electric Co., Cleveland and has been with it ever since. In 1925, he was elected vice-president and in 1938 secretary of that company.

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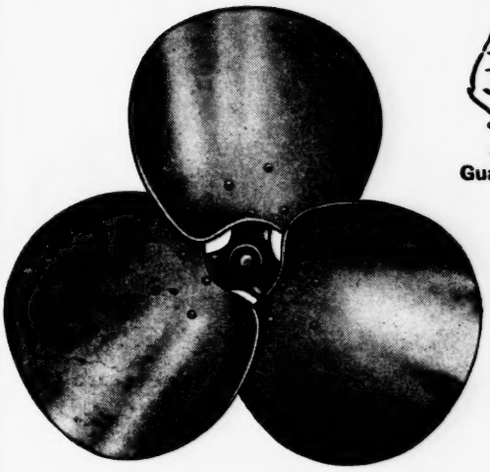

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He is also publisher and editor of "The Stabilizer," a magazine for welders, and several books on welding.

In 1942 he established the A. F. Davis Welding Library at the Ohio State University, said to be the most extensive welding library, and presented it with funds for its continued growth. Mr. Davis is secretary of the James F. Lincoln Arc Welding Foundation that was created in 1936.

E. H. R. Pegg

E. H. R. Pegg *(Dust Control by Combined Centrifugal and Electrostatic Principles*, page 78) received his B.S. in Aeronautical Engineering from the University of Pittsburgh in 1933 and was employed by the Westinghouse Electric Corp. After completion of the Westinghouse graduate student training course in engineering, he was assigned to the materials and process engineering department on various projects involving mechanical, electrical and metallurgical engineering.

In 1937, he was transferred to the new products division and assigned to the development of the new electrostatic air cleaner, later given the trade name "Precipitron". In 1941 he was appointed section engineer in charge of development and application engineering for this product, which position was held until his resignation at the end of 1945 to take a position with the Aerotec Co. of White Plains, New York.

When that company was incorporated early this year, he was appointed chief engineer responsible for development and application of centrifugal and electrical dust collectors for application to industrial and power plant fields.

CANADIAN DEGREE-DAYS FOR MARCH, 1947*

City	March 1947	March Normal	Cumulative, Sept. 1 to Mar. 31 1946-47	Cumulative, Sept. 1 to Mar. 31 Normal
Calgary, Alta.	1339	1215	8279	7881
Charlottetown, P.E.I.	1091	1203	6385	6869
Crescent Valley, B. C.	800	961	6429	6704
Edmonton, Alta.	1404	1290	9314	8589
Fort William, Ont.	1333	1386	8515	8590
Grande Prairie, Alta.	1401	1339	9740	8795
Halifax, N. S.	960	1066	5611	6134
London, Ont.	1166	1073	6046	6326
Medicine Hat, Alta.	1395	1150	8320	7588
Moncton, N. B.	1135	1190	6768	7296
Montreal, P. Q.	1116	1265	6896	7373
North Bay, Ont.	1336	1280	7983	7933
Ottawa, Ont.	1215	1256	7271	7638
Penticton, B. C.	831	787	5839	5566
Porquis Junction, Ont.	1476	1550	8989	9614
Prince George, B. C.	1017	1091	7838	7768
Quebec City, P. Q.	1150	1311	7258	7999
Regina, Sask.	1631	1504	10,104	9641
St. John, N. B.	1011	1132	6105	6784
Saskatoon, Sask.	1600	1445	10,275	9356
Toronto, Ont.	1051	1072	5652	6226
Vancouver, B. C	632	682	4716	4479
Victoria, B. C.	546	651	4171	4065
Windsor, Ont.	1070	995	5476	5872
Winnipeg, Man.	1513	1531	9516	9622

*These data are supplied through the courtesy of the Meteorological Division, Air Services Branch, Department of Transport, Canada, with the cooperation of N. W. Etter, Heating Engineer, Enamel & Heating Products Limited, Sackville, N. B.

The Department reports a slight revision in figures for February as published in last month's issue. Corrected figures are: Penticton, B. C., 893; Prince George, B. C., 1196; Regina, Sask., 1898.

INDUSTRIAL DEGREE-DAYS
March, 1947

City	55F Base	45F Base
Baltimore, Md.	460	182
Buffalo, N. Y.	779	472
Chicago, Ill.	671	367
Cleveland, Ohio	715	414
Detroit, Mich.	748	442
Indianapolis, Ind.	638	345
New York, N. Y.	541	247
Philadelphia, Pa.	498	208
Pittsburgh, Pa.	698	397
St. Louis, Mo.	555	266

. . . Since the Last Issue

D. Boggs

Avery Engineering Co., Cleveland, Ohio, announces the appointment of Dennis Boggs as chief engineer. Mr. Boggs is a graduate of the University of Kentucky, class of 1936, with a degree in electrical engineering. He is a registered Ohio Professional Engineer, member of the ASHVE and of the ASRE. Mr. Boggs joined the Avery Engineering Organization in the Cincinnati office in 1942. His work will include the supervision of all engineering pertaining to refrigeration and air conditioning contracts. The company, distributor in northern Ohio for several lines of refrigeration and air conditioning products, was devoted entirely to industrial refrigeration and air conditioning applications during the war, and is now expanding in straight commercial air conditioning.

After 52 years in the pipe business, **Edwin P. Corey** has resigned as general manager of tubular sales of **The Youngstown Sheet and Tube Co.** effective March 31, 1947. Mr. Corey held this position with Youngstown since October, 1923. Prior to this, he held various positions with the company, having started with them on February 10, 1895.

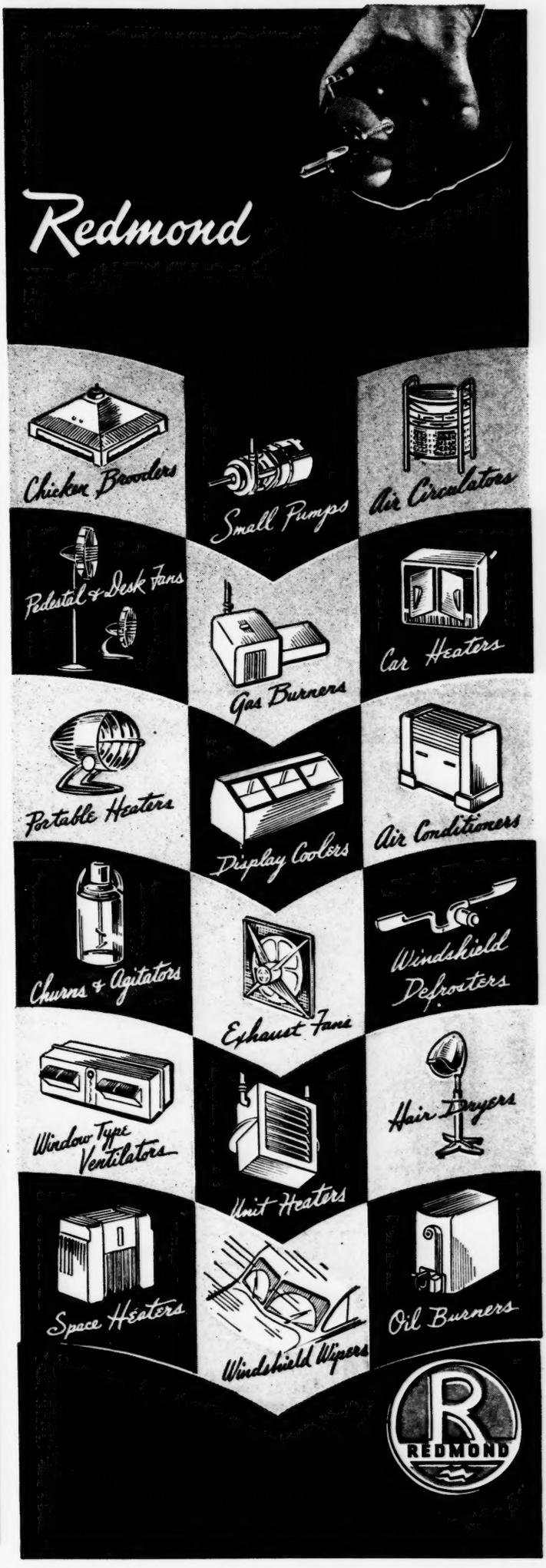
The Chicago offices of **Tube Turns, Inc.**, of Louisville, Ky., have been moved to suite 904 Fairbanks-Morse Building, 600 South Michigan Avenue. The phone number, HArrison 8527, remains the same.

Dudley & Garland, newly formed Boston sales engineering firm has been appointed New England representative of the Industrial Division of **Bryant Heater Co., Cleveland.** The new firm is located at 176 Federal St.

E. P. Murr, 1340 East 14th Ave., Denver, Col., is **Research Products Corp.** representative in Colorado and Wyoming. The company manufactures air filters, water treating minerals, and expanded fiber products.

Air Devices, Inc., recently announced the appointment of **Standard Air and Lite Corp.**, as representative for their comp'ete line of air filters, diffusers, exhausters, and industrial furnaces, in the Pittsburgh area. Their headquarters will be located at 713 Penn Avenue, Pittsburgh 22, Pa.

Alfred J. Yardley has been elected president of **Jenkins Brcs.**, manufacturers of valves, succeeding his father, **Farnham Yardley**, who becomes chairman of the board.



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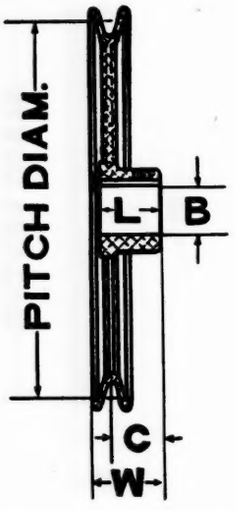
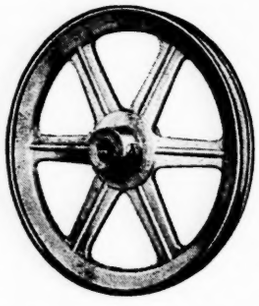
Variable Pitch Pulleys

Silent and true running, diamond bored to plus or minus .0005"—CENTRAL Variable Pitch Pulleys are available to meet all conditions for fractional horsepower drives on HEATING, VENTILATING, AIR CONDITIONING and similar equipment. Speed variations of approximately 30% may be obtained by including a Variable Pitch Pulley in the drive—variations of about 500% by using both pulleys of this type and by interchanging them on the motor and driven shafts. Grooves accurately built to the belt, assuring perfect grip. For following V-Belts: 0—3/8", A—1/2" and B—5/8". Dimensions: Bore diameter, 1/2" to 1". Width of belt groove, min. 1/2", max. 7/8". Overall length of pulley, 1 5/8".

"CENTRAL"
V-Grooved Pulleys





For Fans, Blowers, Stokers, Air Conditioners, Pumps, Refrigerators, Oil Burners, Compressors. Dimensions: *For A—1/2" Belts:* Pitch Dia., 1 1/8" to 13 5/8". Bore Dia., 1/2" to 1 1/2". Bore Length, 1 5/32" to 1 7/16". Overall Width, 1 5/16" to 1 5/8". Center Dist. from face of hub: 15/16" to 1 1/4".

For B—5/8" Belts: Pitch Dia. 1 1/2" to 13 1/2". Bore Dia., 1/2" to 1 1/2". Bore Length, 1 1/4" to 1 17/32". Overall Width, 1 13/32" to 1 23/32". Center Dist. from face of hub, 15/16 to 1 1/4".

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Flexible Couplings

Extreme simplicity of construction cuts assembling costs, eliminates constant adjusting and servicing. Consist of two metal flanges and one non-metallic insert. Available in wide variety of sizes. What are your transmission requirements?

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Walter P. Berg has been elected vice-president and member of the board of directors of **Dravo Corp.**, Pittsburgh, Pa. A graduate of Lehigh University (M.E. degree, 1917), Mr. Berg came to Dravo in 1923 as an assistant engineer. He was active in engineering and sales work until 1938 when he became manager of the power department of the corporation's Machinery Division. In 1946 Mr. Berg was appointed general manager of the Machinery division, in which capacity he continues in addition to his new duties.

W. P. Berg

Appointment of three men to head divisions of the newly organized commercial engineering divisions of **General Electric's** Air Conditioning Department, has been announced. **W. B. Miller** has become engineer of the field engineering division; **C. W. Brown**, engineer of the application engineering division; **C. E. Ehrenhardt**, engineer of the engineering commercial division; and **R. W. Olsen**, supervisor, field section, of the field engineering division.

Appointment of **Huntington Sharp** as sales promotion manager of the **Herman Nelson Corp.**, Moline, Ill., manufacturers of heating and ventilating equipment, has been announced. Mr. Sharp, associated with the firm since September 1944 as manager of the sales control department, has served as a trade and news magazine editor and in public relations and advertising.

Charles S. Reynolds of Utica, New York, is the most recent addition to the Master Kraft oil heating factory field force. He will cover New York upstate from Albany to Buffalo for **Harvey-Whipple, Inc.**

The Youngstown Sheet and Tube Co. announces the appointment, effective April 1, 1947, of **G. W. Christopher** as manager of standard pipe sales and **J. W. Owings** as manager of oil country tubular sales.

Coal Heating Service of Grand Rapids, Inc., activated on April 1st by the **National Coal Association**, brings the total of bituminous coal handled at retail by members of Coal Heating Service groups to approximately 4,534,520 tons.

Young Radiator Co., Racine, Wisconsin, manufacturer of heat transfer equipment, has announced the addition to its staff of **James A. Stahn** as engineering representative.

Mr. Stahn will be connected with aircraft heat transfer products and will also contribute to the engineering, research and production of heat transfer equipment for gas turbines and other contemplated developments.

J. A. Stahn

Graduated from Purdue University in 1941 with the degree of B.S.M.E., he entered the Army Air Forces in September of 1941. Under the AAF training program his formal education was continued at the University of California where he received his M.E. in 1946. Prior to coming to Young Radiator Company he concluded his military service with the rank of Major, AAF.

A. D. Robertson, formerly assistant manager of sales and engineering of the electrical section at the Norwood, Ohio, Works of the **Allis-Chalmers Mfg. Co.**, has been named manager of the company's Tampa, Fla., district office succeeding the late Berrien Moore.

E. M. Flannery, vice-president of the Bush Manufacturing Co., Hartford, Conn., was elected president of the **Refrigeration Equipment Manufacturers Association** at the annual meeting in Chicago, April 9 to 11. Other new officers elected were **H. F. Hildreth**, sales manager, refrigeration specialties division, Westinghouse Electric Corp., Springfield, Mass., vice-president; **R. H. Israel**, sales manager, Virginia Smelting Co., West Norfolk, Va., treasurer, and **K. B. Thorndike**, vice-president, Detroit Lubricator Co., Chicago, secretary. These three, with Mr. Flannery, comprise the association's executive committee for the coming year.

U. S. Radiator Corp., in its annual report for the fiscal year ended January 31, 1947, showed net sales of $17,970,873 and net profit after taxes of $1,193,694. These figures compare with the previous year's net sales of $11,606,017 and net profit of $261,325.

William P. Webster has been elected to the board of directors of **Clark Manufacturing Co.**, producers of fluid controls for industry. He joined the company in 1944 as office manager, later becoming purchasing agent, then assistant to the secretary.

Brison Wood has been elected treasurer of the **Minneapolis-Honeywell Regulator Co.** Mr. Wood succeeds Willard L. Huff, who has been serving as executive vice-president and treasurer. Mr. Huff will continue to serve as a director of the company and executive vice-president. Joining Minneapolis-Honeywell in March of 1942 to assist in expediting war production, Mr. Wood was elected assistant secretary and assistant treasurer the following year, which positions he has held until his election to treasurer. Prior to the war, Mr. Wood was associated with the firm of Coldwell, Banker and Company, real estate brokers of Los Angeles, until he joined the Honeywell organization.

B. Wood

Rapid expansion of home construction and industry in Montana has led to the opening of a Minneapolis-Honeywell office in Butte. **James A. Reynolds**, a native of Montana, has been transferred from the company's Seattle office to serve as branch manager for the new territory.

Daniel B. Benscoter, Cincinnati, has been appointed district sales engineer for the Machinery Division of **Dravo Corp.**, Pittsburgh, which represents DeLaval Steam Turbine Co., and which manufactures and markets Dravo Counterflo commercial and industrial space heaters. His office is in the Southern Ohio Bank Building.

At the eighth annual meeting of the **Propeller Fan Manufacturers' Association** held at New Orleans, Louisiana, **W. J. Lohman, Jr.**, president; **M. L. Aitken**, vice-president; and **L. O. Monroe**, secretary-treasurer. Mr. Lohman is treasurer of Chelsea Products, Inc., Irvington, New Jersey, and Mr. Aitken is general manager of Propellair, Inc., Springfield, Ohio.

Reginald C. Cushing has been appointed sales manager of the air conditioning controls division for the Pacific region of the **Minneapolis-Honeywell Regulator Co.** Mr. Cushing joined Honeywell in 1935 in the installation department of the air conditioning controls division and later was promoted to salesman. He will make his sales headquarters at the company's San Francisco office, 1136 Howard Street.

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Each unit is made to deliver full rated capacity under most difficult conditions — 12,000 BTU's of air conditioning performance for every ton of rating. TYPHOON's exclusive large flo-turn cooling coils and the oversized all-copper condenser, are two of the many features that make for trouble-free performance and maximum customer satisfaction.

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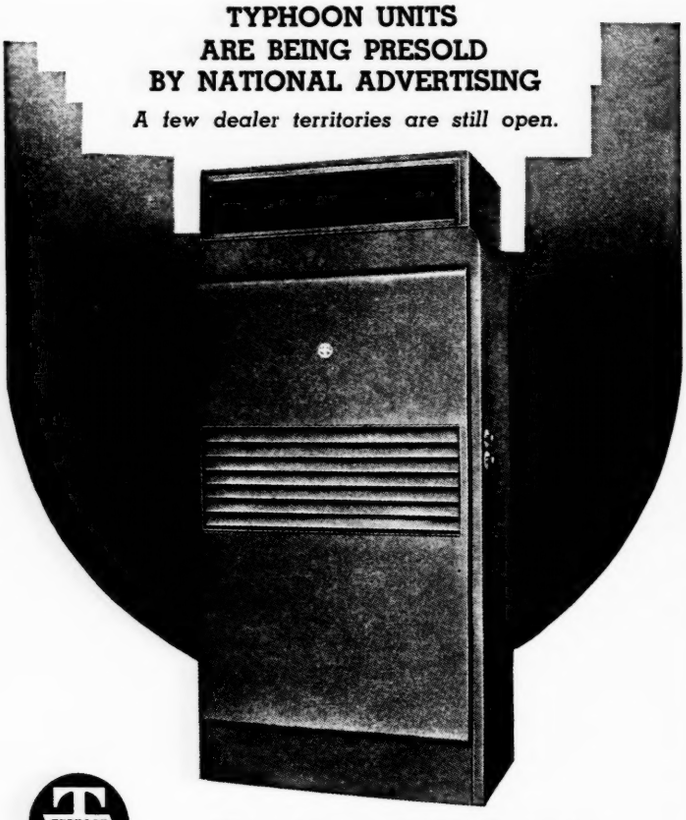
A few dealer territories are still open.



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Division of Ice Air Conditioning Co., Inc.
794 UNION STREET BROOKLYN 15, N. Y.

Five new sales management posts have been created within the commercial refrigeration sales department of **Frigidaire Division of General Motors.** The new managerships increase the commercial sales sections from two to seven. **J. A. Smith** has been promoted to manager of farm and locker market sales. Air conditioning sales is being headed by **C. F. Wood.** New manager of special applications is **G. A. Hayner. G. H. Ewing** has been promoted to manager of case and fixture sales. **L. E. Smith** is the new manager of water and beverage cooler sales.

Westinghouse Electric Appliance Division has announced the formation of the Apartment House and Builder Sales Department to replace the former Home Building Department. W. R. Arbuckle, former head of the Home Building Department, will be manager of the new department.

Arthur Nutting, chief engineer, **American Air Filter Co., Inc.,** and **Howard W. Pound,** air filter sales manager, have been appointed vice-presidents of the company, Mr. Nutting in charge of engineering and Mr. Pound in charge of the Air Filter Division. A native of Warsaw, Indiana, Mr. Pound received his B.S.M.E. from Purdue University in 1921. Prior to his graduation he was a naval aviator in World War I. Mr. Pound has had years of practical ex-

H. W. Pound

A. Nutting

perience in the air filtration field and after occupying the position of sales manager of the Independent Air Filter Co., joined AAF seven years ago as sales manager of the Electro-Matic Division. He was later appointed manager of the Air Filter Sales Division, a post which he filled until his present appointment. Born in Louisville, Kentucky, Mr. Nutting, a graduate of the University of Kentucky, received his B.S.M.E. in 1927. Following his graduation, he served a short time with the State Highway Commission before becoming affiliated with AAF in 1928. Since 1937, when he was made chief engineer, Mr. Nutting has been in charge of the American Air Filter Co.'s engineering and research programs.

Official dedication of the **National Warm Air Heating and Air Conditioning Association's** new research residence at Champaign, Illinois, will be witnessed by Association members attending their annual mid-year convention, June 5 and 6. The residence will be a part of the Small Homes Center of the University of Illinois and will replace the Association's first research residence built in 1924.

Donald H. Jensen has joined the **Petroleum Heat and Power Co.** to organize expansion of the company's employe, consumer and public information program. Until recently Mr. Jensen was an account executive with James W. Irwin and Associates.

The part played by air conditioning in the modernization of stores will be one of the principal features of the first annual **Store Modernization Show** to be held at Grand Central Palace, New York, July 7-12, it was announced by John W. H. Evans, managing director of the exposition.

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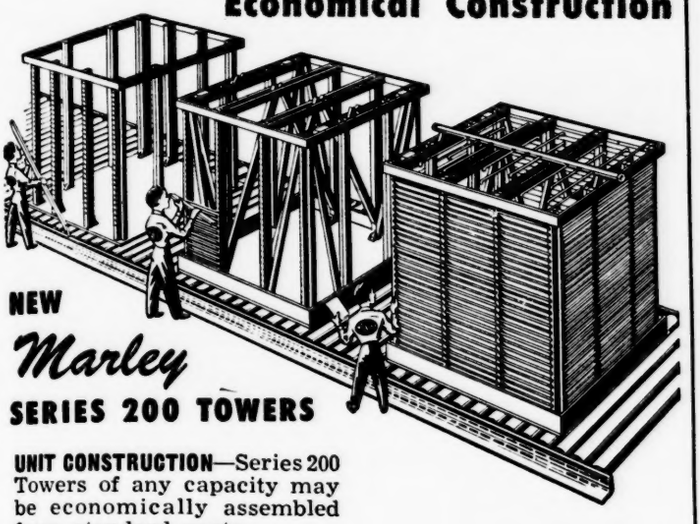
for Volume Production
Economical Construction



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ADDED STRENGTH—Series 200 Towers are framed structurally complete before louver posts and louvers are added. Completely shop fabricated.

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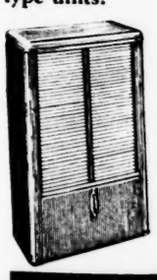

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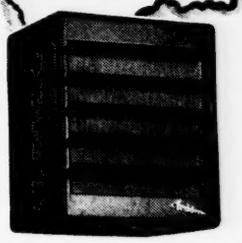

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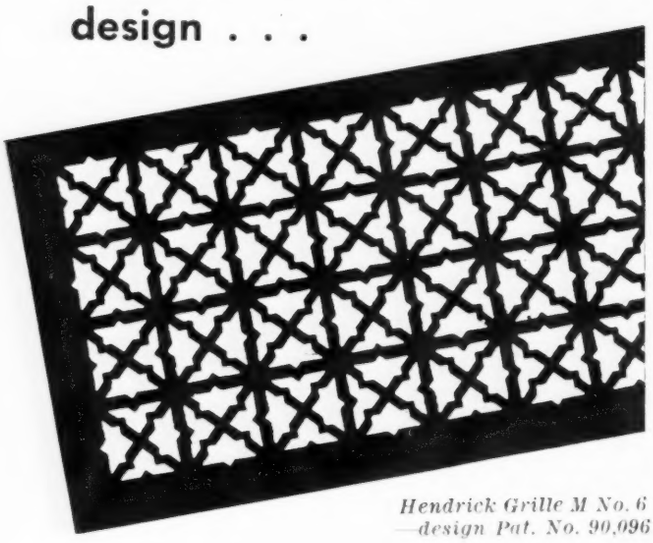

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CONVENTION PROGRAMS

Spring Meeting — ASRE

The spring meeting of the American Society of Refrigerating Engineers will be held at the Alexandria Hotel, Los Angeles, June 9-11. The technical part of the convention program follows:

Monday, June 9

10:00 a.m.—*First Session* — R. H. Money, president of A.S.R.E., presiding

"Refrigeration in Motion Picture Laboratories"—Arthur J. Hess, Hess, Greiner and Polland, Los Angeles, Calif.

"Mechanism of Heat Transfer, Panel Cooling and Heat Storage" — C. S. Leopold, Consulting Engineer, Philadelphia, Pa.

"Surface Temperatures and Heat Flow for a Concrete Slab with an Imbedded Grid of Pipes" — Carl F. Kayan, Columbia University, New York, N. Y.

1:00 p.m.—Welcome Luncheon (speaker to be announced)

Tuesday, June 10

9:30 a.m.—*Second Session*—Three papers dealing with the freezing of foods

12:30 p.m.—Publications Committee luncheon - meeting Group Luncheons, to be arranged as required

2:00 p.m.—*Third Session* — Burgess H. Jennings, vice-president of A.S.R.E., presiding

"Use of Refrigeration and Air Conditioning in the Growing of Plants" — F. W. Went, University of California, Pasadena, Calif.

"Biological Considerations in Hydrocooling and Precooling"—W. T. Pentzer, Plant Physiologist, U. S. Dept. of Agriculture, Fresno, Calif.

"Engineering Considerations in Hydrocooling and Precooling"—James C. Rear, General Manager of Plants, Union Ice Co., San Francisco, Calif.

Wednesday, June 11

9:30 a.m.—*Fourth Session*—Paul B. Christensen, treasurer of A.S.R.E., presiding

"Water Treatment and the Control of Scaling in Evaporative Condensers"—Ralph M. Westcott, Los Angeles, Calif.

"Air Flow Measurement in the Laboratory"—Daniel D. Wile, Development Department, Carrier Corp., Syracuse, N. Y.

"New Developments in Refrigerant Lubricants" — Edgar S. Ross, Development Division, Sun Oil Co., Philadelphia, Pa.

ASHVE Spring Meeting

Three technical sessions have been arranged for the semi-annual meeting of the ASHVE to be held at the Hotel del Coronado, Coronado, Calif., June 2-4. The papers are to be presented at morning sessions starting at 9:30 a.m. The afternoons and evenings have been reserved for entertainment features.

Monday, June 2

Challenge of the ASHVE Research Program, by Cyril Tasker

The Thermo-Physical Properties of Building Materials, by H. B. Nottage

Heat Transfer by Forced Convection Along a Smooth Flat Surface, by G. V. Parmalee and R. G. Huebscher

Influence of Gaseous Radiation in Panel Heating, by F. W. Hutchinson.

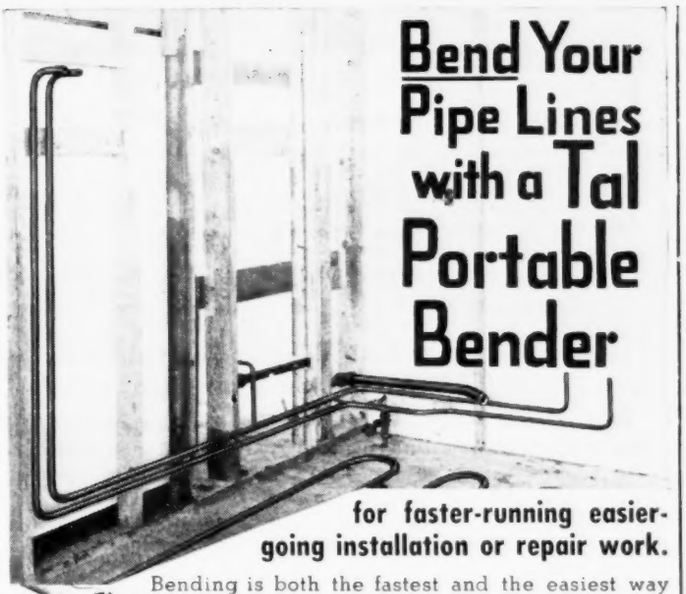
Tuesday, June 3

Conditions for Comfort, by C. S. Leopold

A Method for Improving the Effective Temperature Index, by C. P. Yaglou

Physiological Adjustment of Human Beings to Sudden Change in Environment, by Nathaniel Glickman, Torhru Inouye, Stanley E. Telser, M. D., Robert W. Keeton, M. D., Ford K. Hick, M. D., and Maurice K. Fahnestock

Comfort Reactions of 275 Workers During Occupancy of Air



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31,290 lbs.—.040 x 10¼" x coils—3SO
50,432 lbs.—.040 x 11⅜" x coils—3S ¼ Hard
102,653 lbs.—.040 x 32" x coils—3SO
120,938 lbs.—.040 x 48" x 144"—3SO
49,000 lbs.—.051 x 20" x coils—3S ½ Hard
49,524 lbs.—.051 x 36" x 120"—3SO
109,413 lbs.—.064 x 48" x 144"—3SO
150,000 lbs.—.064 x coils 1" to 12" wide—24SO Alclad
20,000 lbs.—.091 x 3" x coils—24SO Alclad
61,334 lbs.—.091 x 60" x 165"—24ST Alclad
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31,072 lbs.—.125 x 48" x 144"—24SO Alclad
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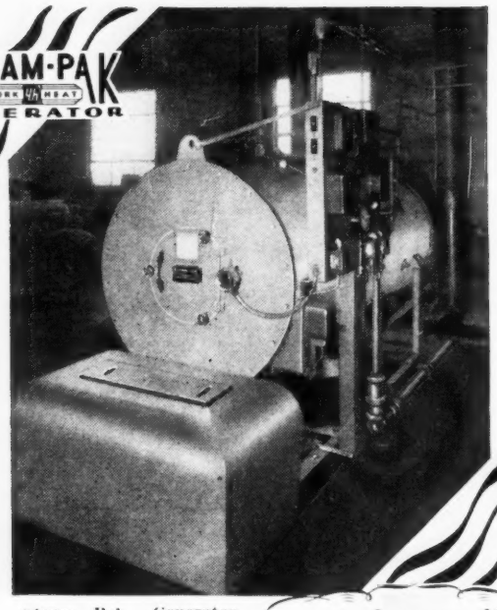

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Steam-Pak Generators are available in high or low pressure models developing 15, 30, 50, 75, or 100 hp. They are designed for use of light or heavy fuel oils (Nos. 3, 5, or 6). Industrial Division, York-Shipley, Inc., York, Pa.

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Conditioned Offices, by Frank B. Rowley, Richard C. Jordan, and Warren E. Snyder

Wednesday, June 4

Experimental Studies on Panel Heating Tube Spacing, by B. F. Raber and F. W. Hutchinson

Effect of Floor Slab on Building Structure Temperatures and Heat Flow, by Carl F. Kayan

Triethylene Glycol Vapor Distribution for Air Sterilization, by Edward Bigg, B. H. Jennings, and F. C. W. Olson

Winter and Summer Design Temperature, by W. L. Holliday

Heating, Piping and Air Conditioning Contractors National Association Convention

Plans have been completed for the program for the 58th annual convention of the Heating, Piping and Air Conditioning Contractors National Association, to be held at the Netherland Plaza, Cincinnati, Ohio, June 11-14. Wednesday, the opening day, will be devoted largely to association business. Dr. N. R. Whitney, economist, The Proctor and Gamble Co., will speak on Outlook for Business. The program for Thursday and Friday follows, with Thursday afternoon devoted to a panel discussion on radiant heating.

Thursday, June 12

9:30 a.m.—*Open Session*

Report on Plumbing and Heating Industries Bureau—Arnold H. Goelz

Report on Committee of Ten—Coal and Heating Industries—Harry M. Hart

Report of Membership Committee—Joseph A. Courter, General Chairman

Report of Committee on Standards—Paul A. Edwards, Chairman

Address—

Report of Boiler Output Committee—Harry M. Hart, Chairman

Report on American Standards Association—Harry M. Hart

Report of Committee on Air Conditioning—A. C. Buensod, Chairman

Report of Committee on Apprenticeship—Arnold H. Goelz, Chairman

2:00 p.m.—*This session will be devoted to home heating and panel heating*

Report of Committee on Trade Promotion—George P. Nachman, Chairman

Report of Home Heating Committee—J. A. Ljunggren, Chairman

Address—

The "Why" and "How" of Panel Heating—A discussion by experts who will answer questions from the floor. Peter B. Gordon, Chairman of Radiant Panel Heating Committee, presiding.

"Temperature Controls in Panel Heating Installations"—John Haines, Vice-President, Minneapolis-Honeywell Regulator Co.

"Experiences with Panel Heating from Contractor's Viewpoint"—George P. Nachman

Question and Answer Period

Friday, June 13

9:30 a.m.—*Open Session*

Report of Commitee on Certification—Wray M. Scott, Chairman

Report of Committee on Welding—Richard S. Tobin, Chairman

Report of National Certified Pipe Welding Bureau—William W. Murray, Jr., Chairman

Report of Editorial Committee, Standard Manual on Pipe Welding—John H. Zink, Chairman

Address—

Report of Secretaries' Conference—Louis T. Braun, Chairman

Saturday morning is reserved for an executive session to wind up the convention business and for an open session at which the new officers will be installed. Association secretary is Joseph C. Fitts, 1250 Avenue of the Americas, New York City.

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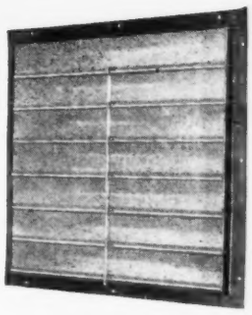

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Actual photo-micrograph of section of Henry sintered disc (magnified 50 diam.)

HENRY FILTERS AND FILTER-DRIERS with new heavy duty *sintered discs*

. . . more efficient
. . . free flow of refrigerant
. . . will not collapse if clogged

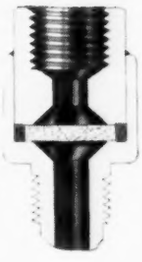
HENRY now offers small filters and filter-driers with sintered metal filtering elements. The sintered metals, in disc form, are made from metallic powders carefully screened for controlled particle size, fused under hydraulic pressure and controlled heat into a porous, homogeneous mass.

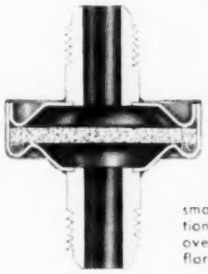
The Henry sintered discs will filter out particles larger than 75 microns (.003") diameter, roughly the filtering equivalent of a 200 mesh screen. While more effectively collecting foreign particles in a system, these discs possess an inherent porosity that assures free passage of the refrigerant. The most important advantage is the fact that Henry sintered discs will not collapse if they become clogged because they are designed to withstand 300 pounds pressure. Ask your Henry jobber for full information on the economies as well as the improved protection these new Henry filters and filter-driers afford.

Special designs can be furnished for volume requirements.

TYPE SF 12

A small unit for domestic refrigerators, water coolers, etc. May be installed directly on expansion valve, etc. 1½" long with ¼" fem. x ¼" male flare.





TYPE SF 21

A larger filter for small commercial installations. 2" diameter, 2¼" overall length. ¼" male flare connections.

TYPE SFD 11

Combination Sintered Filter-Drier. Small efficient unit filled with silica gel. 1¼" diameter shell with ¼" male flare connections.

Other Filter-Driers With Sintered Discs Having 6 and 12 Cubic Inches Drier Capacity Also Available. Ask For Special Bulletin.

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Control Devices, Valves, Driers, Strainers and Accessories for Refrigeration and Air Conditioning and Industrial Applications

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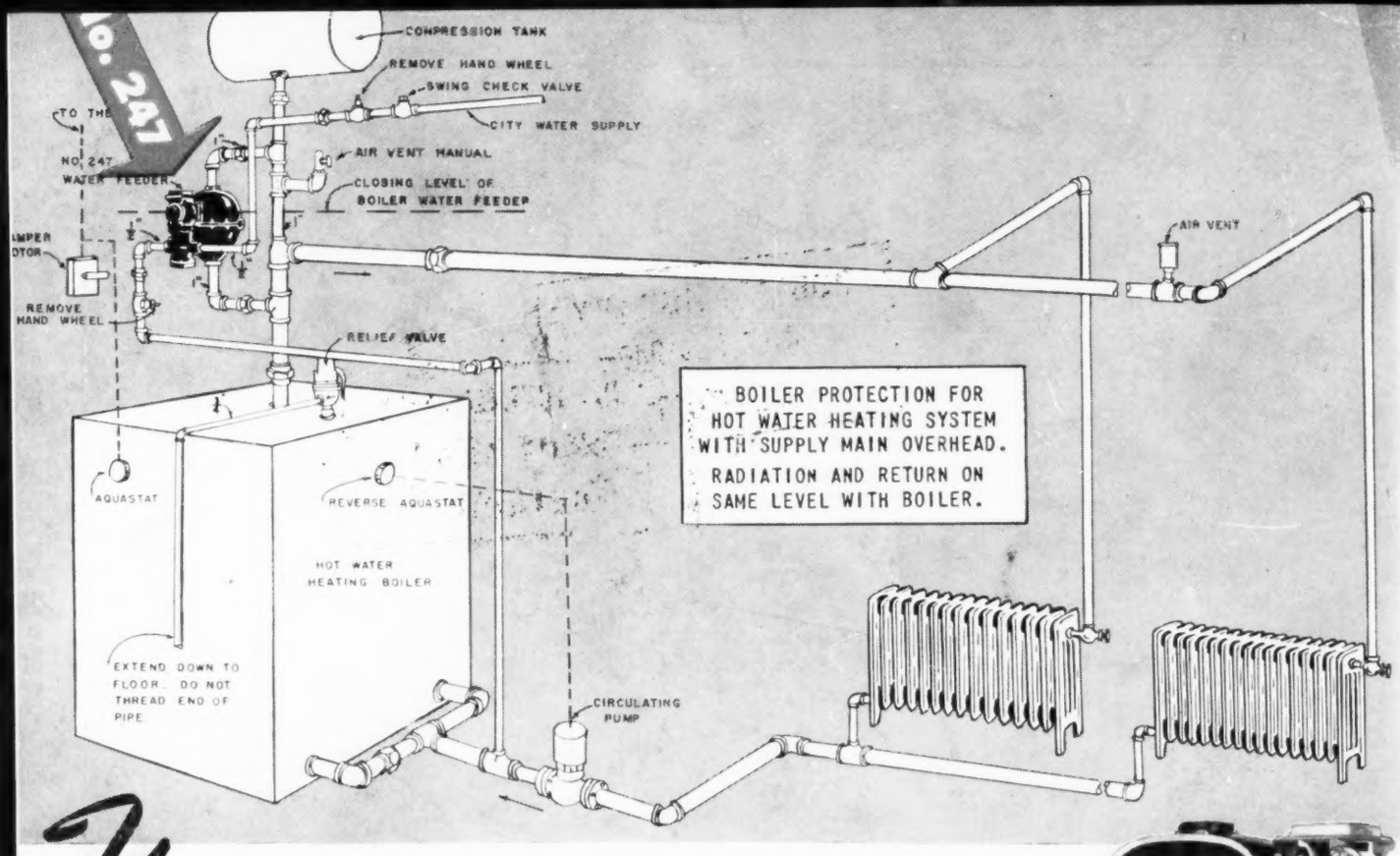
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REMOVE HAND WHEEL
SWING CHECK VALVE
CITY WATER SUPPLY
AIR VENT MANUAL
CLOSING LEVEL OF BOILER WATER FEEDER
AIR VENT
RELIEF VALVE
AQUASTAT
REVERSE AQUASTAT
HOT WATER HEATING BOILER
EXTEND DOWN TO FLOOR. DO NOT THREAD END OF PIPE
CIRCULATING PUMP
BOILER PROTECTION FOR HOT WATER HEATING SYSTEM WITH SUPPLY MAIN OVERHEAD. RADIATION AND RETURN ON SAME LEVEL WITH BOILER.

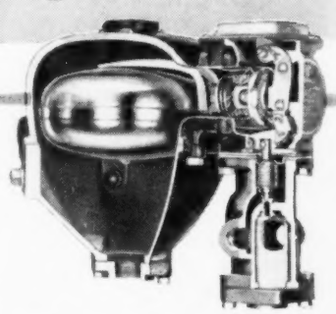
Yes, boiler water feeders on hot water heating systems!

• Some old timers may shrug their shoulders a little at the sight of water feeders on hot water heating boilers. The fact remains, however, that there will be more and more of them. They are being written into many hot water specifications; recommended for certain types of hot water systems by leading consulting engineers.

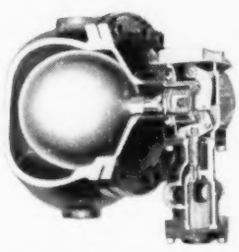
A typical application of a float operated boiler water feeder on a hot water system now being widely used is shown in the schematic drawing above—a forced circulation system with radiation on the same level as the boiler. We first ran into this need for a float operated boiler water feeder while studying the application of our Btu. rated safety relief valves. In army camps and housing projects where this and similar systems were used during the war, it was found that merely holding down pressure by means of an adequate relief valve could not, under certain conditions, prevent damage and boiler losses.

As a result, we were granted permission to conduct a series of experiments on systems of this type at Camp Carson, Colorado. In the course of this work it was found that the trouble could be corrected by installing a float operated boiler water feeder as shown in the drawing to keep the system filled to its highest point. And so conclusive were the results that the hook-up became standard under "Corps of Engineers Specification CE-51 Forced Hot Water Heating Systems," for systems of this kind. Many float operated feeders have since been installed on these "same-level" systems and *there is no case on record of any trouble with plants so equipped.*

In applying float operated boiler water feeders to hot water systems it has also been found that in addition to the protection of the boiler, a number of gains are made in the efficiency and dependability of operation of the system. The story is too involved to be told in limited space, but many eye-opening and helpful facts are available to those interested in this rapidly broadening field.



McDonnell No. 247 boiler water feeder used on type of hot water system shown above. It is the same as the McDonnell No. 47 except that it is for installation with equalizing piping. It is also available with low water cut-off switch (as No. 247-2) for automatically fired hot water boilers.



McDonnell No. 51 boiler water feeder for the larger boilers. This type of feeder is also available with low water cut-off switch (as No. 51-2) for automatically fired hot water boilers.

No. 150 low water control for boilers of any size with pressures up to 150 lbs. The No. 150 may be used as low water cut-off; low water alarm, or as a combination of these functions.

McDONNELL & MILLER, INC., 1313 Wrigley Bldg., Chicago 11, Ill.

Your Jobber Stocks Them

Doing One Thing Well

McDONNELL *Boiler Water Level* CONTROLS

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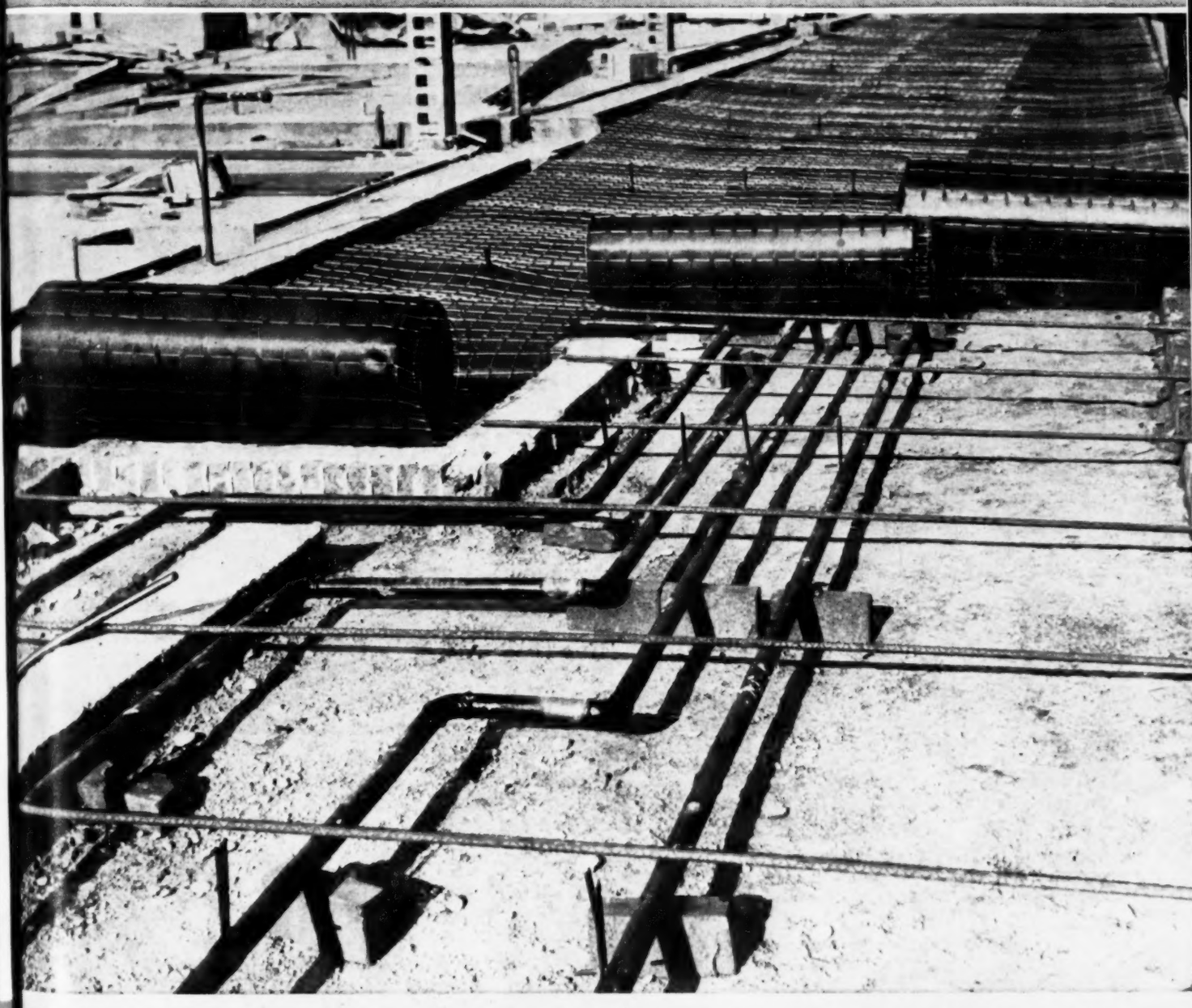

This sidewalk in front of the Union Bus Terminal in Pittsburgh will be kept clear in winter by a snow-melting system of iron pipe enclosed in a six-inch air space under the concrete and carrying steam. Heat loss to the ground will be slowed by layers of slag and a thin layer of concrete, while transfer to the concrete walk will be facilitated by metal-to-metal contact between pipes, cross rods, and reinforcing steel mesh woven into the heavy retaining paper which forms the "ceiling" of the air space. Photo by A. M. Byers Co.

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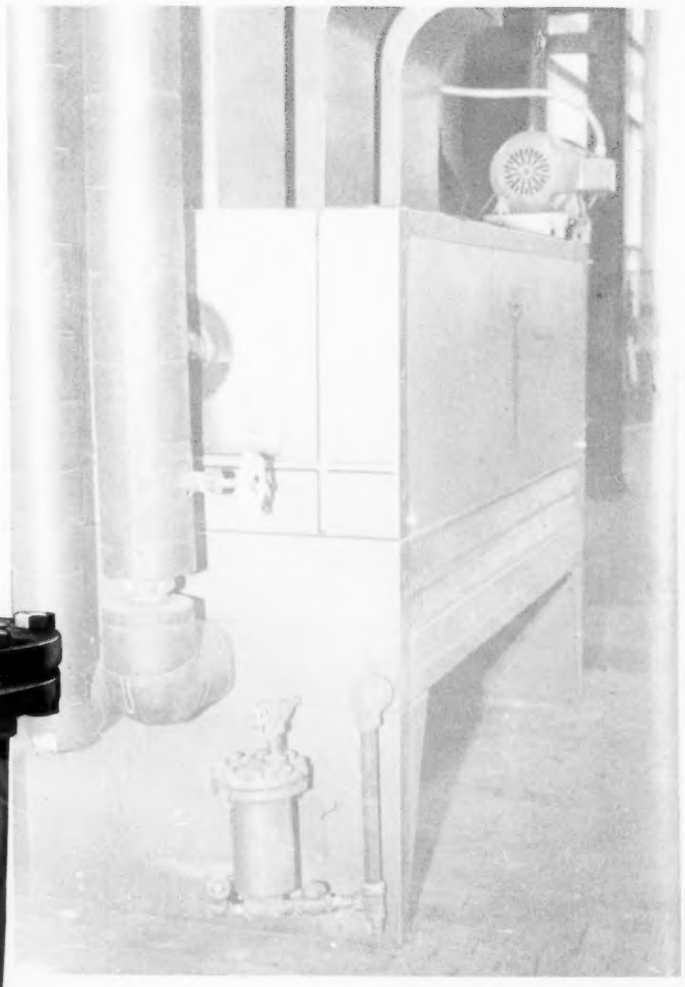
GOOD NEWS

for Heating Engineers and Contractors



— A new, low cost ALL-STEEL steam trap

QUICK FACTS

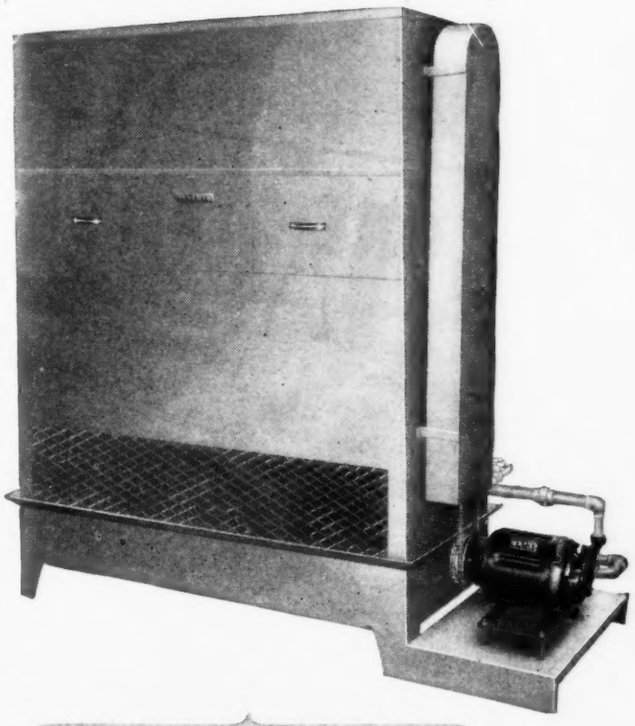
- SMALL SIZE—Height, 7"; diameter, 4¼". ½" pipe connections.
- BIG CAPACITY—Up to 900 lbs. per hour continuous discharge capacity, depending on pressure.
- PRESSURE—Suitable for operating pressures up to 250 p.s.i. Body is actually built to withstand much higher pressures.
- ORDERING—List price, $14.00. Specify operating pressure.

THE NEW Armstrong No. 3211 *forged steel* trap is designed and built to meet the needs of heating engineers and contractors for a small size, moderately priced, high quality steam trap. *It's tough!* — the forged steel body will withstand hard knocks that would shatter ordinary cast iron traps. *It's fire-resistant!* — no danger of the body weakening under high pressure when exposed to fire. Inside, it has the same *all-stainless steel* operating mechanism that's used in Armstrong traps built for pressures up to 1500 p.s.i.!

This new trap is now in production in the Armstrong factory. If you are in need of traps of this type, see your local Armstrong representative at once. If it is an urgent matter, write, phone, or wire ARMSTRONG MACHINE WORKS, 846 Maple St., Three Rivers, Michigan.

ARMSTRONG *Inverted Bucket* STEAM TRAPS

Over a Million in Use... For Power... Process... Heating



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Evaporative Condensers

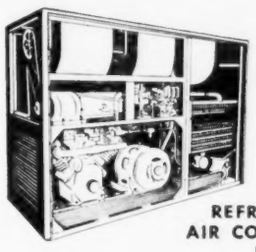
More Efficient Condensing of Refrigerants PLUS Big Water Savings . . .

The evaporative condenser is a "packaged" unit in that it incorporates all the desirable features of the conventional shell and tube condenser, water cooling tower and water pump in a single, compact piece of equipment.

Savings up to 95% in water costs are possible with the use of evaporative condensers for refrigeration or air conditioning applications. They also eliminate waste-water disposal problems, and pumping costs are much lower than those of a conventional cooling tower installation.

usAIRco Evaporative Condensers are designed for maximum economy and efficiency in operation. An unusually generous amount of coil surface, made up of all prime condenser surface, plus ratings for low condensing temperatures to minimize scaling, provide highest condensing efficiency. Blowers are accurately balanced for quiet operation; forwardly-curved blade design moves larger volume of air at less power cost.

Complete design and engineering information may be had on usAIRco Evaporative Condensers by requesting a copy of special bulletin.



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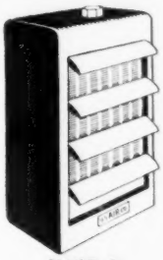

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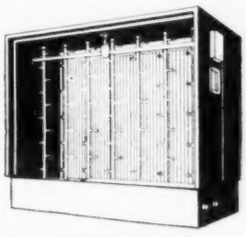

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